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HEAT EXCHANGER DEVELOPMENT FOR WASTE WATER HEAT RECOVERY

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Thesis

HEAT EXCHANGER DEVELOPMENT
FOR
WASTE WATER HEAT RECOVERY

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Abstract

Hot water plays an import role in modern life. The consumption of hot water represents a significant part of the nation's energy consumption. One way of reducing the energy consumption involved, and hence the cost of that energy, is to reclaim heat from the waste warm water that is discharged to the sewer each day.

The potential for economic waste water heat recovery depends on both the quantity available and whether the quality fits the requirement of the heating load. To recover heat from waste water in residential and commercial buildings is hard to achieve in quality because of its low temperature range. Nevertheless, efforts to recycle this waste energy could result in significant energy savings.

The objective of this research was to develop a multiple panel thermosyphon heat exchanger for a waste water heat recovery system. The advantage of the system proposed in this work is that it not only provides useful energy transfer during simultaneous flow of cold supply and warm drain water but also has the ability to store recovered energy at the bottom of a hot water storage tank for later use. While this concept is not new, the design of the heat exchanger proposed for the present study is significantly different from those used previously.

Component experiments were carried out to determine the performance characteristics of a single thermosyphon panel. By changing the inclination angle of the single panel heat exchanger and varying its working condition, it was found that the inclination angle of 10° could be identified as the minimum inclination angle at which good performance was still obtained. The close values of the overall heat transfer coefficients between top surface of the panel insulated and both top and bottom surfaces of the panel uninsulated shows that the overall heat transfer coefficient of the single panel was dominated by the bottom surface of the panel. Even if in a worst case the top surface of the panel might be possibly covered by the deposits from the waste water, it would not affect much on the heat transfer performance of the panel.

Measurements of hot water usage and waste water temperature and flow rates were obtained for a potential application of the proposed exchanger (the dishwasher for the kitchen in the University Halls of Residence).

A model of a multi-panel thermosyphon heat exchanger was also developed to predict the energy savings that would be expected if such a heat exchanger was used in this situation. The result indicated that an overall electricity of 7500 kWh could be saved annually from the dishwasher system by employing a four-panel thermosyphon heat exchanger.

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NOMENCLATURE

Symbol	Description	Units
A	heat transfer surface area	m^2
$c_{p,w}$, $c_{p,c}$	specific heat at constant pressure	J/kg.K
$\dot{E}_{in}$	rate of energy transfer in heat exchanger	W
$\dot{E}_{out}$	rate of energy transfer out of heat exchanger	W
F	LMTD correction factor	
$\bar{h}$	overall convection heat transfer coefficient	$W/m^2.K$
k	thermal conductivity	W/m.K
$\dot{m}_c$	mass flow rate of the cold fluid	kg/s
$\dot{m}_w$	mass flow rate of the hot fluid	kg/s
n	number of thermosyphon panels	
NTU	number of transfer units	
$\dot{q}$	heat transfer rate	W
$\dot{q}_{loss}$	heat loss from the surface of heat exchanger tank to ambient air	W
t	time interval of data input	second
$T_{c,in}$	temperature of the cold fluid entering heat exchanger	K
$T_{c,out}$	temperature of the cold fluid out of heat exchanger	K
$T_{w,in}$	temperature of the hot fluid entering heat exchanger	K
$T_{w,out}$	temperature of the hot fluid out of heat exchanger	K
$\bar{U}$	overall heat transfer coefficient	$W/m^2.K$,
V	volume	m^3
ΔT_m	log mean temperature difference	K
c_p	specific heat	J/kg.K
ε	heat exchanger effectiveness	
ρ	mass density	kg/m^3

Chapter 1: INTRODUCTION

1.1 Background

Hot water plays an import role in modern life. We each use between 40 and 60 litres of hot water per day for bathing and washing. A sizable restaurant or cafeteria may use thousands of litres of hot water for dishwashing every day[1]. The consumption of hot water represents a significant part of the nation's energy consumption. Statistics shows that about 14% of the country's total energy and 35% of all electricity is used domestically, with about 38% of domestic energy consumption being used for water heating [1] (See Figure 1.1-1). In typical non-residential building, domestic hot water creates about 4% of the annual energy consumption. In buildings where sleeping or food preparation occur, including hotels, restaurants, and hospitals, domestic hot water may account for as much as 30% of total energy assumption [2].

For domestic demand, hot water usage influenced by household activities is varied from 160 litres for low use to 330 litres for high use each day for a 4-person family[3]. Typical cold water supply temperatures range from 10°C to 20°C. Since the temperature from the storage tank is usually about 60°C, the temperature rise required ranges from 40K to 50K and the energy requirement can therefore vary by as much as 25% for the same supply service in terms of hot water delivery.

The amount of energy used for water heating in a typical New Zealand household is usually quoted between 3000 and 7000 kWh/yr with an average of about 3500 to 4500 kWh/yr. The national average may increase with the increasing use of dishwashers and washing machines and larger water cylinders. The national annual energy consumption for hot water is estimated to be about 5,500 GWh for electric cylinders [4].

[1] See Chapter 4 for a recorded daily hot water usage of a dishwasher in the cafeteria of University Halls of Residence, UCSA.

The domestic market for water heaters is quite large. Each year there are about 50,000 water systems in existing houses being upgraded and around 20,000 new homes being built, without mentioning the renovations of water heating systems in commercial and institutional buildings.

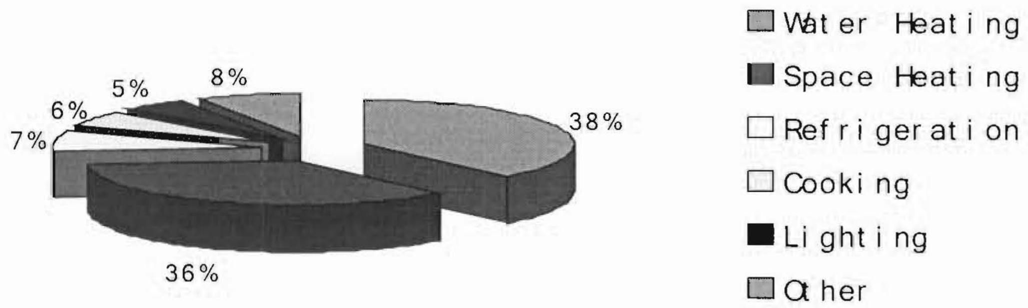


Figure 1.1-1: Domestic energy uses[2]

1.2 Heat Recovery System

In recent years, considerable attention is being given with the greatly increasing demand for domestic hot water and cost of fuel that produces the heat. One way of reducing the energy consumption involved, and hence the cost of that energy, is to reclaim heat from the waste warm water that is discharged to the sewer each day.

Depending on how much hot water is "diluted" by the addition of cold water to give a desired usage temperature, the waste water is directed down the drain at a temperature of 35~55°C. Considering that water for hot supply is heated from typically 10~20°C to 60~70°C by the water heater, it is apparent that approximately 70~80% of all hot water energy literally goes down the drain, carrying with it up to 5250 MWh of energy a day in New Zealand alone.

The potential for economic waste heat recovery depends on both the quantity available and whether the quality fits the requirement of the heating load. To recover heat from waste water in residential and commercial buildings is hard to achieve in

[2] Energy Management Handbook, by Wayne C. Turner. Third Edition. 1997. The Fairmont Press, Inc., USA.

quality because of its low temperature range. However if only we could transfer 25~30% of the heat from the warm waste water into the cold supply water we could save ourselves the cost of a considerable proportion of the energy that is used in the heating process. Efforts to recycle this waste energy could result in significant energy savings.

From the consumer's point of view, the principal reason for attempting to recover waste heat from the waste warm water is economic. The use of waste heat recovery can improve the energy efficiency and reduce the consumption and the cost of the energy. It most be noted from the outset, however, that the true economics cannot look at the energy cost savings alone: the capital and installation cost of any energy-saving device must be considered as well, and the energy cost savings must provide an acceptable rate of return on the investment made.

From a national energy supply point of view, power demand is expected to grow from around 30,000GWh in 1995 to around 45,000 GWh or more by 2020 [5]. After experiencing two serious power shortages in the past three dry years, most New Zealanders are aware that there's some sort of crisis occurring in the electricity industry in this country. Some of plans are being considered such as building some coal-fired, gas-fired, hydro, wind-powered stations in an effort to increase the country power supply. Meridian Energy had proposed a $1.5 billion Project Aqua scheme on the Waitaki River, involving construction of six new hydro power stations on a 60km canal, with an intake at Kurow and an outfall 6km from the coast. However, it was abandoned recently because of the environmental impact it would have on the surrounding area. Any future plans for generating power will take several years to implement and need a large amount of investment. One of the alternatives to this is to increase energy efficiency by recovering waste heat so as to obtain 'extra' energy.

1.3 Previous Development of Heat Recovery System

Due to increasing use of residential hot water use and the potential economic benefit in heat recovery, more and more companies and researchers are trying to find a way to recover waste heat from showering, bathing and dishwashing. Some waste water heat recovery systems have been designed with the employment of different types of heat exchangers

The first example of practical applications is called 'GFX Drain Water Heat Recovery System', invented by Carmine Vasile from America in the late 1970s. The GFX (Gravity Film Exchanger) works on the principle that heat transfers from warm water through pipe and tube walls to cold water. Warm waste shower water travels down the walls of a vertical 3-inch-diameter (75mm) drain pipe in a thin 0.012 to 0.027 mm film which transfers heat rapidly to the pipe. Cold water that passes through a standard 1/2-inch (12.5mm) spiral copper tube which wraps tightly around the pipe intends to capture that heat from the pipe (See Figure 1.3-1). GFX claims an average of 34% energy saving with a payback of 1.6-4.6 years [6]. This system has advantages of simple installation and compact structure; however its efficiency is largely dependent on the contact surface area between drain pipe and tube.

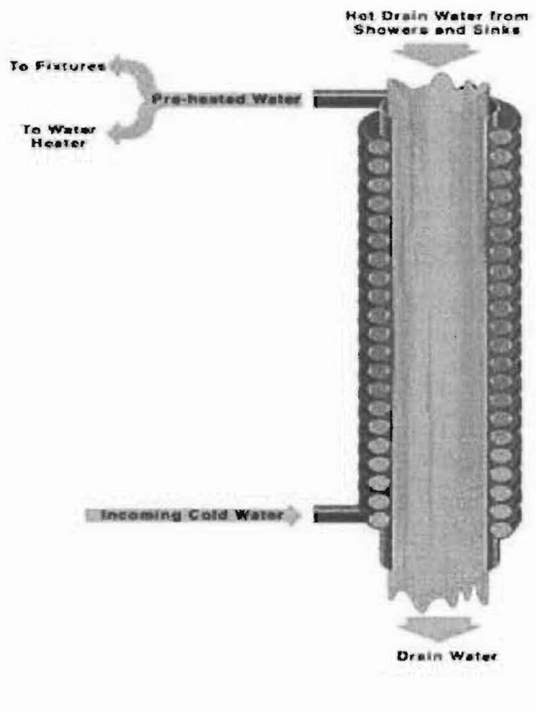


Figure 1.3-1: GFX drain water heat recovery system[3]

[3] Obtained from website oikos.com

Another waste water heat recover system developed by a Dutch company, Gastec NV has two concentric pipes. The inner drain pipe transfers heat through to the outer pipe with incoming cold water in a counter flow. The system is simple but rather long (2.5m) and comparatively expensive. And it also has no double walls to protect cold supply water from the possibility of contamination caused by pipe leakage.

In 1970's, I. E. Smith [7] from Cranfield Institute of Technology, UK carried out an experimental study on a heat recovery system installed in a house which showed a 10 per cent of energy saving in the total energy consumption. In his system, a small storage tank was located inside the bigger waste collection tank. The ascending cold water running in the small tank was preheated by the descending hot waste water in the big tank. Because suspended material in the waste water can freely pass through the outlet valve, an advantage of this system was low maintenance due to its simplicity in structure.

In late 1980's, G. J. Parker and Dr. A.S. Tucker from University of Canterbury designed and dynamically simulated a domestic hot water system which included a waste water heat exchanger and/or a solar panel collector. The heat exchanger was a concentric cylinder unit. A 70-litre cylinder containing cold water to be pre-heated was surrounded by an annular space of 160 litres of waste warm water. Three tests were carried to study the effects of three thermostat settings for the storage tank on the energy use and water quantity for three water usage patterns (low, average and high usage patterns). The tests included Basic System test; Basic system plus waste-water heat exchanger; Basic system plus heat exchanger plus solar panels. The research shows that the energy saved by only employing the heat exchanger reached a maximum of 32% [8].

In 1993, Dr. D.M. Clucas from Mechanical Department of University of Canterbury developed a heat recovery system specifically for showers. The concept of the system was that a shower tray installed on the floor of a shower cabinet carried warm water from the shower and an approximately 15m long copper pipe with flowing cold water was attached to the underside of the tray to absorb heat. The system was simple in design and could be easily produced. But it also brings discomfort to the shower-user because of cooling of the shower tray, making it necessary to use a plastic mat as a layer of insulation.

Most recently in New Zealand, an Auckland student developed a shower heat recovery unit under the supervision of Dr. Joe Deans from University of Auckland [9]. The device includes coiled pipes immersed in a chamber containing the waste water, which flows down and into an inner cylinder from which it exits into the drain. The unit raises the cold water temperature from 18°C to 35°C and has an estimated $200 of annual electricity savings each year. The system is simply structured and has impressive energy efficiency when the fouling effect is not considered. But the long term efficiency will reduce with the increase of fouling resistance. Besides, there is likelihood of water contamination if leakage occurs in the coiled cold water tube.

1.4 Proposed Heat Recovery System

A practical and efficient way for the wastewater heat recovery is to position a heat exchanger in the waste pipeline and remove the heat continuously from the waste not only as it is discharged but between discharges as well. The energy obtained preheats the incoming circulating cold supply water before it is heated by the heating element in hot water cylinder (See Figure 1.4-1). The advantage of the system is that it not only provides useful energy during simultaneous flow of cold supply and warm drain water but also has the ability to store recovered heat at the bottom of a hot water storage tank for later use. Whole this concept is not new [7], the design of the heat exchanger proposed for the present study is significantly different from those used previously. The rationale behind this new concept is explained in Chapter 2.

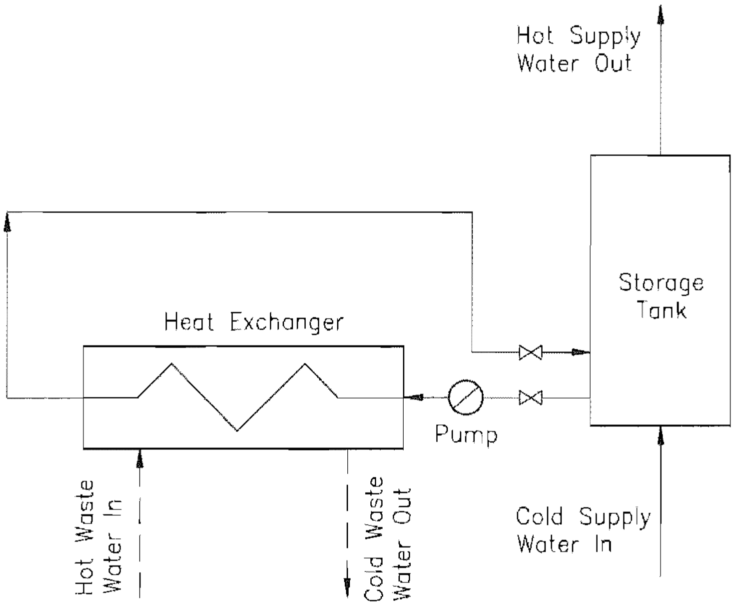


Figure 1.4-1: Illustration of waster water heat recovery system

1.5 Objectives of the Research

1). To implement a thorough literature study on thermosyphon and waste heat recovery systems, and explore the possibility of introducing a thermosyphon heat exchanger for the development of waste water heat recovery. This part will be discussed in Chapter 2.

2). To design a simple heat exchanger which consists of a single thermosyphon 'Heatsheet' and carry out a series of laboratory based experiments which will lead to a full understanding of thermosyphon heat exchanger performance. This will be introduced in Chapter 3.

3). To choose a waste heat recovery system which has a representative hot water usage pattern to which the thermosyphon heat exchanger might be applied for practical use. Some typical field data will be collected for further thermosyphon heat exchanger development. The analysis of the chosen heat recovery will be presented in Chapter 4.

4). To build up a thermosyphon heat exchanger model that has a set of 'Heatsheet' panels installed in series and programme in the waster water heat recovery system. The proposed heat exchanger design would utilize two important principles: thermal stratification to minimise irreversible mixing of cold and hot waste water discharges; and a heat pipe system to minimise the potential for cross-contamination between the two fluids streams but without interposing significant thermal resistance. This model will be introduced in Chapter 5.

5). To analyse the thermosyphon heat exchanger performance and optimise the heat exchanger and its heat recovery system to explore the economic feasibility of the research. This will be presented in Chapter 6.

6). The pros and cons of the thermosyphon heat exchanger will be discussed in the last Chapter 7 to draw a conclusion on the research currently done. Suggestions and recommendations for further study will be given.

Chapter 2: DEVELOPMENT OF HEAT EXCHANGER CONCEPT

In energy conservation, recovery and utilization, heat exchangers play a crucial and dominant role. Many new and innovative heat exchangers have been developed for waste heat recovery and utilization of renewable energy. Also, many heat exchangers have been refined and made more reliable and more efficient.

Considering the heat transfer process, heat exchangers may be classified into five types, which are direct transfer, storage, fluidized bed, direct contact and fired types[10]. Direct transfer heat exchangers might be further categorised into conventional and unconventional heat exchangers in terms of their established usage and construction features. The one in which heat is transferred between two working fluids that flow through the device is a called conventional heat exchanger. One example of conventional heat exchangers is the shell and tube heat exchanger. The unconventional heat exchanger is an alternative heat exchanger which often contains two or more intermediate fluids that flow in separate working areas through the device. Therefore the heat transfer in an unconventional heat exchanger is more complicated than a conventional heat exchanger. Thermosyphons and heat pipes are two examples of unconventional heat exchangers.

2.1 Shell and Tube Heat Exchanger

One of most commonly used direct heat exchangers in liquid-liquid heat recovery is a shell-and-tube heat exchanger. The heat exchanger surface consists of a number of tubes, spaced apart, with one fluid flowing through the tubes and the other fluid outside the tubes. The ends of each tube are joined to corresponding holes in tube sheet(s), being rolled or/and welded to the tube sheet(s). The tubes are generally kept in position on the outside by supporting plates or/and cross-baffles (See Figure 2.1-1).

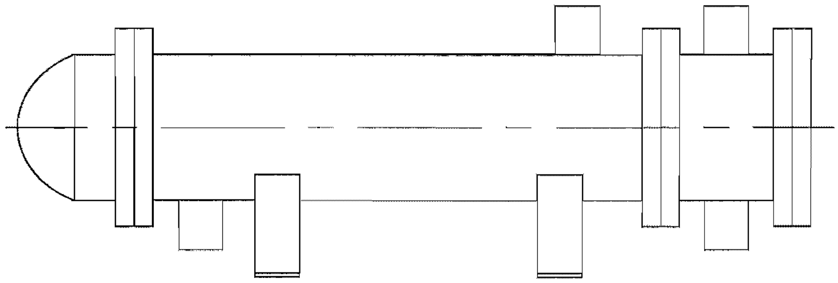


Figure 2.1-1: Shell and tube heat exchanger

2.1.1 Advantages of Shell & Tube heat exchanger

The conventional shell-and-tube heat exchanger is commonly produced in large quantities. It is made of available standard materials and the manufacturing technique is well known. One of the significant advantages for a shell-and-tube heat exchanger is its ability to withstand very high pressures and temperatures. It could be designed to be mounted on the ground, vertically mounted or even inclined to suit different situations.

The shell-and-tube heat exchanger has a variety of types and sizes to choose from. It can also be designed by changing the number of tubes, type of tube arrays and the size of each tube to achieve any capacity. The heat transfer surface area available per unit volume of the exchanger ranges from 100 to 500 m^2/m^3 [10].

2.1.2 Disadvantages of Shell & Tube heat exchanger

Because the shell-and-tube heat exchanger has a tube bundle fixed by support plates, baffles and tube sheet(s), the shell side of the heat exchanger can be very hard to maintain. It is usually recommended for clean fluids, only, or a dirty fluid flowing on the inside of the tubes and clean fluid on the outside, provided the tube side can be cleaned regularly. For any waste water heat recovery system, inevitably one of the fluids (the waste water) cannot be regarded as being "clean". When fluids are not so “clean” and the flow rate of the fluids is varied and small, the expensive shell-and-tube heat exchanger is definitely not suitable from an economic point of view.

Another serious problem for a conventional heat exchanger proposed to be used in waste heat recovery is the likelihood of fresh water being contaminated by waste water once a tube is damaged due to the leakage of itself or welding joints between tubes and tube sheet. The single wall structure might be an advantage for effective heat transfer. However, it is unable to delay or prevent waste water from getting through leaks and into fresh water. Double wall tubing would be an expensive way of guarding against this possibility and, from a heat transfer point of view, would introduce additional thermal resistance between the hot and cold fluids.

As the driving potential of heat transfer is temperature gradient, if at any time the waste water is colder than the supply water which is supposed to be heated, sophisticated control of the waste water circuit would be required for a conventional heat exchanger to avoid the counter-productive transfer of heat from warm fresh water to cold waste water.

Since there is a likelihood of leakage in conventional heat exchangers during their life, catastrophic consequences might result in the hot water supply being contaminated by waste water. As a result of these disadvantages, a new type of heat exchanger incorporating a thermosyphon system is considered. The operating principles of thermosyphons are described next.

2.2 Thermosyphon system

2.2.1 Historical Development of Thermosyphon System

The thermosyphon is similar in many respects to a heat pipe and may be considered as a sub-set of heat pipes. Perkins Tube, a hot water hermetic heating tube invented by A.M. Perkins [11] for his concentric tube boiler in 1839, has been regarded as an initial part of the history of thermosyphon.

F.W. Gay modified Perkins Tube in his device in 1929. In his patent, a number of finned Perkins Tubes or thermosyphon tubes were arranged as in the conventional gas/gas heat pipe heat exchanger, with the evaporator sections located vertically below the condensers, a plate sealing the passage between the exhaust and inlet air duct. Working fluids include methanol, water and mercury, depending on the likely exhaust gas temperatures [12].

In 1942, the heat pipe concept was first suggested by R. S. Gaugler [13] for cooling the interior of an icebox. The heat pipe was employed to transfer heat from the interior compartment of the refrigerator to a pan located below the compartment and containing crushed ice.

However, its real potential was not realized for 20 years until it was rediscovered by Grover [14] who coined the name "heat pipe". Since then its remarkable properties became appreciated and serious development work took place. Thermosyphon system theory was developed by P.D.Dunn and D.A.Reay in their book "Heat Pipes"[12] in 1970s, and has been widely used, especially in the heat recovery field since then.

2.2.2 Characteristics of Thermosyphon system

An illustration of a thermosyphon is shown in Figure 2.2-1. A heat pipe is a closed tube or vessel which has been evacuated, partially filled with a heat transfer fluid, and permanently sealed at the ends. When the lower end of the tube is heated, the liquid vaporizes and the vapour moves to the cold end of the tube where it is condensed. The condensate is returned to the hot end by gravity. Since the latent heat of evaporation is large, considerable quantities of heat can be transported with a very small temperature difference from end to end, operating essentially uniformly at saturation temperature corresponding to the pressure within the tube. Thus the structure will have a high effective thermal conductance.
The thermosyphon is characterized by [12]:

1) Very high effective thermal conductance: This is probably its most important characteristics. Because of the small temperature drop for vapour flow, a simple

thermosyphon could have a high thermal conductivity as to enable heat to be transferred over long distance.

2) The ability to act as a thermal flux transformer: Having different surface areas for the evaporator and condenser sections, it can act as a thermal transformer since $q''_{evap}/q''_{cond} = A_{cond}/A_{evap}$.

3) The ability to act as a thermal diode: When acting as a thermal diode, the thermosyphon permits heat to flow in one direction only. Heat is allowed to flow from the evaporator to the condenser when the temperature at the evaporator end is higher than that at the condenser end. As the temperature gradient is reversed, the thermosyphon will stop working. This characteristic is useful in heat recovery applications and will be discussed later in this chapter.

4) As isothermal surface of low thermal impedance. Since the thermosyphon is made up of a condenser and an evaporator, the temperature of the working fluid is essentially constant within the thermosyphon. The condenser surface of thermosyphon will tend to operate at uniform temperature.

5) Flexibility and constructional simplicity. The basic structure of a thermosyphon consists of a container and vapour space. It can take on a variety of shapes to find its way into a particular application.

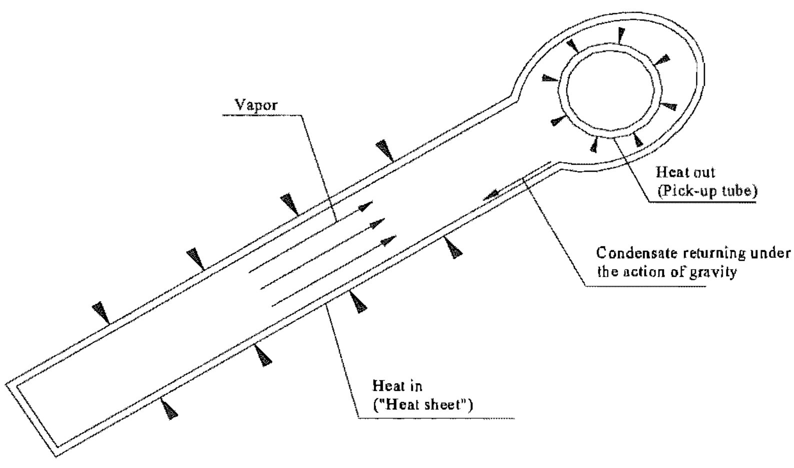


Figure 2.2-1: Illustration of a thermosyphon

2.3 The "Heatsheet" thermosyphon system

The thermosyphon "Heatsheet" [15] operates on the same principle but differs radically in its geometry (See Figure 2.3-1). It consists of two dimpled metal plates with their edges being welded together. The "Heatsheet" is then evacuated and partially filled with a working liquid. The regions of contact between two dimpled plates provide areas for small pools of liquid to accumulate. During operation heat is transferred from the both upper and lower surfaces of the sheet to those pools. Evaporation occurs from the liquid surface, carrying the latent heat of vaporization of the liquid upwards to the top of the sheets where the heat is picked up by a cold water supply tube wrapped in the sheet. The condensate then falls by gravity along the inner walls of sheet back down and forms the liquid pools again. While the "Heatsheet" was conceived as a form of flat plate water heater, its internal working principle may also be applied to heat exchange situations in which the energy source is non-solar.

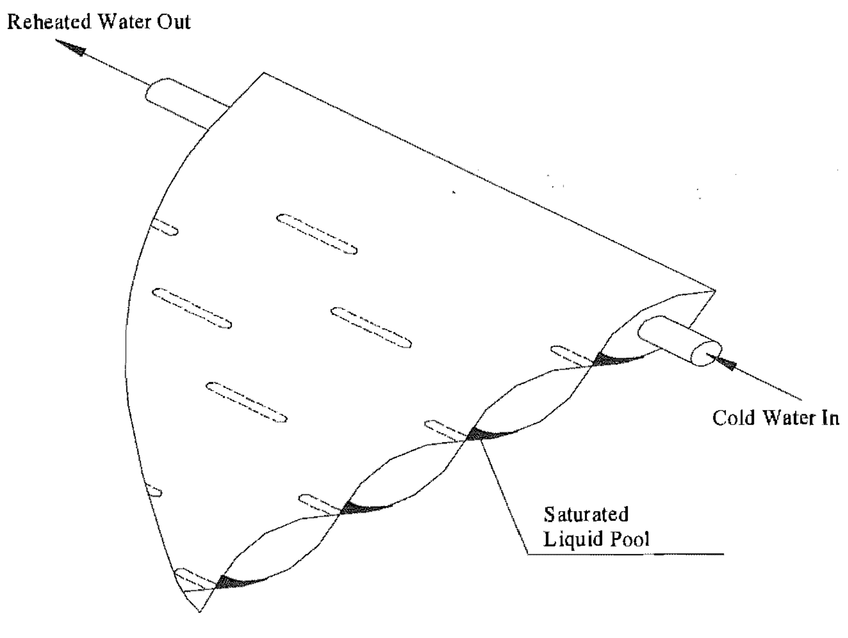


Figure 2.3-1: Schematic of "Heatsheet"

2.4 Thermosyphon Heat Exchanger

The proposed thermosyphon heat exchanger consists of a set of "Heatsheets" in series (See Figure 2.4-1), each of which is constructed of dimpled steel plates and transfers heat internally by two-phase thermosyphon process that takes the place of the conventional tube-and-shell structure (See Figure 2.1-1). The "Heatsheets" are placed at a downward slope in parallel in a holding tank. When warm waste water enters the tank, it is thermally stratified due to the temperature gradient. The thermal energy of waste water is transferred to the "Heatsheet" surface. The sufficiently large heat transfer surface makes it possible to remove considerable amount of heat to the working fluid inside the "Heatsheet". The cold fresh water tube at the top takes away heat from the vaporized liquid as it condenses.

The isothermal characteristic of the thermosyphon heat exchanger means that using a number of "Heatsheets" in series provides a large surface area to transfer more heat. In this heat exchanger, each "Heatsheet" works individually in counter flow, absorbing heat from the waste water and bringing the cold water outlet temperature up one step on a temperature "stairway". A set of "Heatsheets" in series will increase the cold water outlet temperature closer to the saturation temperature of the working fluid of each step and have a better approximation to counter flow.

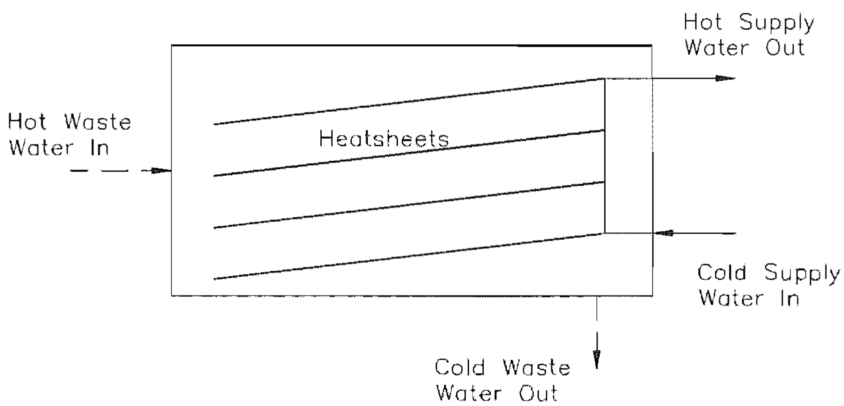


Figure 2.4-1: Proposed thermosyphon heat exchanger

2.5 Selection of Working Fluid in Heat Sheet

From the Water Authorities' point of view, "the water heat must be designed and constructed such that leakage during its life is unlikely. This covers all aspects of materials, manufacture, installation and maintenance" [16].

Some limitations should taken into account in order to determine the most acceptable working fluid.

1). Non-toxic. It is based on the consideration of the likelihood of working fluid contaminating the water supply in case of leakage of the working fluid into the water tube.

2). Compatibility with absorber materials. To avoid corrosion damage to the 'Heatsheet' and degradation of the fluid, it should not react with the sheet.

3). Good thermal stability. For long life operation, it is a necessary feature of the working fluid to keep good thermal properties over its operating temperature range.

4). High latent heat. As a heat carrier, a high latent of vaporization for the working fluid is desirable to allow transfer of large amounts of heat with a minimum fluid flow.

5). Low cost. The fluid should be easily available. An acceptable cost of working fluid is also decisive factor that cannot be overlooked in the practical use of the heat exchanger.

Consideration narrowed the range of likely working fluids to those shown in Table 2.5-1 [17]:

Table 2.5-1: Potential working fluid for the panels

Name	Boiling Point @ 1 atm	Latent Heat kJ/mole
Toluene(C_7H_8)	110.63	33.18
Water(H_2O)	100.0	40.65
Benzene(C_6H_6)	80.09	30.72
Ethanol(C_2H_5OH)	78.29	38.56
Hexane(C_6H_{14})	68.73	28.85
Methanol(CH_3OH)	64.6	35.21
Acetone(C_3H_5OH)	56.05	29.10

2.6 Feasibility of thermosyphon Heat Exchanger

The shortcomings of a conventional tube-shell heat exchanger largely correspond to the advantages that a thermosyphon heat exchanger shows us. The features of thermosyphon heat exchangers that make them attractive for heat recovery applications are similar to those of the thermosyphon and are as follows:

1) No moving parts and no external power requirements for the "Heatsheets", insuring high reliability and minimum maintenance. Compared to a conventional one, the thermosyphon heat exchanger also has the advantages of simplicity in construction, cheap manufacturing cost and relative high rates of heat transfer.

2) Cross contamination between hot and cold fluids is essentially eliminated since a sealed space separates the two flows. The supply water pipe is wrapped and sealed in the 'Heatsheet' which prevents it from contacting waste water directly and minimizes the possibility of waste water contaminating supply water due to the failure of pipe.

3) The thermosyphon heat exchanger permits heat to transfer in one direction, which is from hot waste water to cold fresh water only. The heat transfer can be switched off automatically by the thermosyphon and therefore heat loss is cut down, in the circumstances of the waste water being colder than the supply water in pipe.

As discussed in later Chapter 3, a thermosyphon is influenced in its performance by the angle of operation, since gravity plays in important role in aiding or resisting return flow of the condensate. Because of this sensitivity, the pumping power and ultimately the heat transport may be controlled by tilting the exchanger. For the proposed thermosyphon heat exchanger, the heat sheets could be adjusted to a certain slope as a mean of controlling the heat transfer.

When a thermosyphon "Heatsheet" is flat or slightly sloped, the regions of contact between the dimples on the "Heatsheet" and the flat sheet provides more area for more pools of liquid to accumulate. The smaller thermal resistance of liquid compared to that of vapour results in a high heat transfer between hot liquid and cold working fluid. It would be desirable to have the "Heatsheets" as close to horizontal as possible so the working fluid could have more contact area with the bottom sheet for a better heat transfer performance.

Besides, according to previous research [18], the overall thermal resistance of the thermosyphon is very sensitive to the operating pressure, the heat flux and the quantity of the working fluid. This brings some difficulties in the selection of working fluid and pressure.

This factors, and the lack of previous tests on "Heatsheet" type panels being used as liquid-liquid heat exchangers, meant that there was a clear need to carry out a range of performance tests on a panel of this type.

Chapter 3: THERMAL PERFORMANCE FOR SINGLE PANEL

3.1 Introduction of Thermal Performance Calculation

3.1.1 Temperature distribution of Single Thermosyphon Panel

There are three fluids existing in a heat exchanger which is based on a single thermosyphon panel. The hotter and cooler fluids leave in the same liquid phase state while the intervening internal working fluid experiences phase change during the heat transfer. The result is that, as heat is transferred from hotter fluid to the intermediate fluid then to the cooler fluid, the hotter fluid temperature diminishes and the cooler fluid temperature rises while the intermediate fluid temperature remains unchanged.

Figure 3.1-1 illustrates the temperature changes that occur in the fluids along a single thermosyphon panel heat exchanger. Evidently, it is possible for the outlet temperature of hot and cold fluids to approach closely the saturated temperature of intermediate fluid. The ideal that can achieve is that the two outlet temperatures may be close together in the case of parallel-flow exchanger. As the outlet temperature of the cold fluid can only be as high as the saturated temperature of intermediate fluid, the intermediate fluid will be critical for the thermosyphon heat exchanger. If the saturation line of intermediate fluid moves upward, closer to the upper line, the outlet temperature of the lower line will follow up correspondingly. The final selection of intermediate fluid from the potential "candidate" fluids listed in Table 2.5-1 will be discussed later in this chapter.

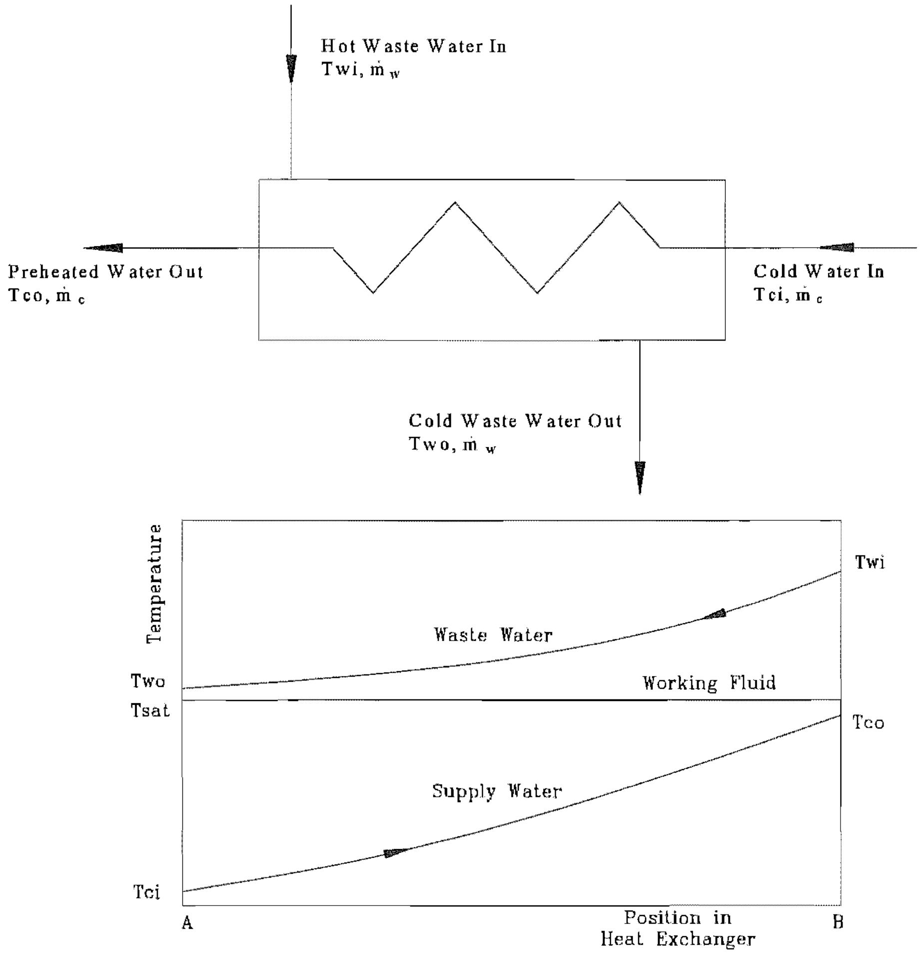


Figure 3.1-1: Temperature distribution for a single thermosyphon heat exchanger

3.1.2 Energy Conservation Analysis

The law of energy conservation says, that at any instant of time the amounts of energy inflow and internal generation (which act to increase the amount of energy stored within the control volume), less energy outflow (which act to decrease the stored energy) represents the rate of change of energy in the control volume.

$$\dot{E}_{in} + \dot{E}_g - \dot{E}_{out} = \frac{dE_{st}}{dt} = \dot{E}_{st} \qquad \text{(Equation 3-1)}$$

If $\dot{q}$ is the total rate of heat transfer between the hot and the cold fluids and heat transfer between the exchanger and its surroundings is negligible, as well as negligible potential and kinetic energy changes, the heat which enters the heat exchanger will be taken out of the heat exchanger instantly. When fluids are not undergoing a phase change and constant specific heats are assumed, the applying overall energy balance equations are:

$$\dot{E}_{in} = \dot{E}_{out} = \dot{q} \qquad \text{(Equation 3-2)}$$

$$\dot{E}_{in} = \dot{m}_w c_{p,w} (T_{w,in} - T_{w,out}) \qquad \text{(Equation 3-3)}$$

$$\dot{E}_{out} = \dot{m}_c c_{p,c} (T_{c,o} - T_{c,in}) + \dot{q}_{loss} \qquad \text{(Equation 3-4)}$$

where $\dot{E}_{in}$ and $\dot{E}_{out}$ are the rates of energy transfer in and out of heat exchanger, W,

$\dot{m}_w$ and $\dot{m}_c$, mass flow rates of the hot and the cold fluids, kg/s,

$c_{p,w}$ and $c_{p,c}$, specific heat at constant pressure, J/kg.K,

$T_{w,in}$ and $T_{w,out}$, temperatures of the hot fluid in and out of heat exchanger, K,

$T_{c,in}$ and $T_{c,out}$, temperatures of the cold fluid in and out of heat exchanger, K,

$\dot{q}_{loss}$, heat loss from the surface of heat exchanger tank to ambient air, W.

3.1.3 The LMTD Method

It is simple matter to use the log mean temperature difference (LMTD) method to evaluate overall the heat transfer coefficient when inlet and outlet temperatures for both fluids are known.

The overall heat transfer rate between the two fluids is:

$$\dot{q} = \overline{U} A F \Delta T_m \qquad \text{(Equation 3-5)}$$

where $\overline{U}$ is overall heat transfer coefficient, $W/m^2.K$,

A, heat transfer surface area, m^2,

F, LMTD correction factor, $F \leq 1$, (F=1 for simple parallel or counter flow)

$\Delta T_m = \dfrac{\Delta T_1 - \Delta T_2}{\ln(\Delta n_1 / \Delta \Delta_2)}$, is log mean temperature difference, K.

$\Delta T_1 = T_{w,in} - T_{c,in}$, $\Delta T_2 = T_{w,out} - T_{c,out}$, K

$\dot{q}$ can be obtained by using equations Eq.3.2, Eq.3.3 and Eq.3.4 when the mass flow rate of either the hot or the cold fluid is specified.

3.2 Experimental Set-Up for a Single Panel Performance

3.2.1 Objectives

The thermal heat transfer rate of a single panel can be expressed in the form of a linear performance characteristic relating mass flow rate and temperature difference between inlet and outlet of the fluid $\dot{q} \propto \dot{m}(T_{in} - T_{out})$. For a panel if the values of $\dot{m}$, Tin and Tout are known, the panel heat performance will hence be identified.

A laboratory experiment was carried out by installing a single panel in a well-insulated tank and supplying it with fluids at well-controlled flow rates and inlet temperatures. The prime purpose of this experiment was to set up a testing prototype heat exchanger to study the performance of the single component panel under different steady state operating conditions. As the performance of single panelled heat exchanger is strongly influenced by the way in which it is installed and operated, the test so devised should also lead to an indication of possible improvement in performance either from design or installation at the next stage.

3.2.2 Consideration of Varying Factors

3.2.2.1 Ambient Air Temperature

As the air temperature changes so there is a change in the amount of heat lost from the heat exchanger to air. The heat exchanger heat loss is directly proportional to the temperature difference between the average temperature of hot water inside the heat exchanger and the ambient temperature, which is room temperature under laboratory conditions. As the laboratory in which the experiment was undertaken has a cooling fan running in summer, the room temperature remained reasonably stable at about 20°C. The tank is well insulated with 30mm thick polystyrene which has high thermal resistance to minimise heat loss. Hence the heat loss of the heat exchanger is small and would not affect the heat transfer performance more than a negligible amount.

3.2.2.2 Cold Supply Water Inlet Temperature and Flowrate

As shown in Figure 3.1-1 and Equations 3.3 & 3.4, if the volume of hot water is much larger than the volume of cold water, the hot water temperature change would be comparatively small. The hot water temperature over the panel could be assumed constant. Since heat transfer rate is proportional to $\overline{U}\Delta T_m$ (See Eq.3-5), an increase of the cold water inlet temperature will result in the decrease of ΔT_m hence a reduction of the heat transfer rate, assuming the overall heat transfer coefficient $\overline{U}$ remains stable.

Increasing the cold water flow rate can be expected to result in an increase in the overall heat transfer coefficient $\overline{U}$ and hence the heat transfer rate because of the increasing Reynolds number.

3.2.2.3 Hot Water Inlet temperature and Flowrate

A higher hot water inlet temperature will widen the temperature difference between hot and coldwater temperatures and therefore increase the heat transfer rate. Because the heat exchange tank was designed to hold a large volume of hot water, the hot water flows at a rather low speed. We could assume essentially free convection occurs between the hot water and the surface of the panel. The hot water flowrate might have an effect on heat exchanger performance but not greatly. This has to be confirmed in the experiment.

3.2.2.4 Inclination Angle of Single Panel Heat Exchanger

Theoretically, a two-phase thermosyphon is driven the gravity acting on the working fluid inside the panel; the panel should be able to operate at any angle as long as working fluid is able to vaporize and condense.

However, in a sloped single panel heat exchanger, when waste water from a small pipe enters the comparatively large heat exchanger, the flow velocity is reduced sufficiently that we might consider that free convection occurs between fluid and the surface of heat sheet. Fluid motion is due to buoyancy forces within the fluid. Buoyancy is due to the combined presence of a fluid density gradient and a body force, where the former is due to temperature gradient and the latter is gravitational (See Figure 3.2-1).

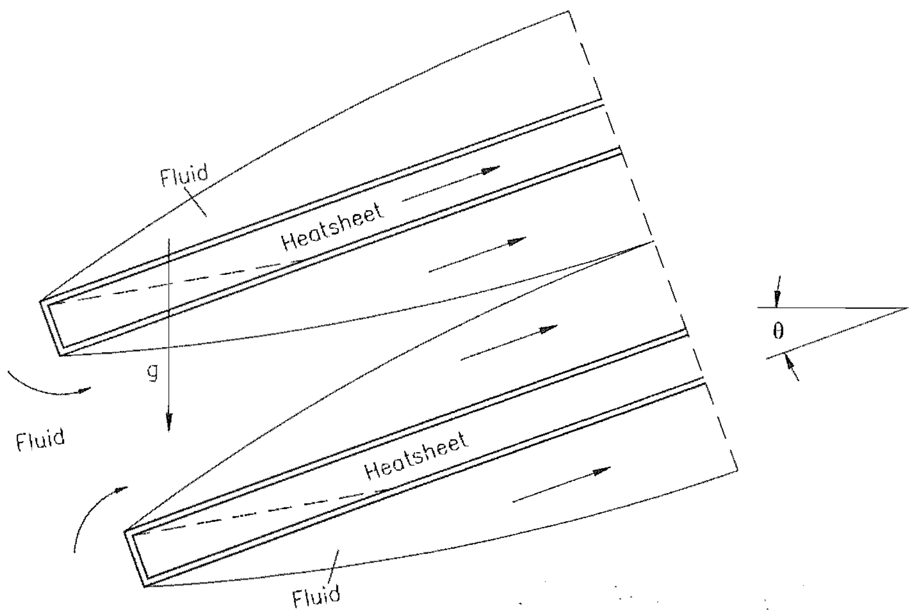


Figure 3.2-1: Schematic of free convection flow between heated parallel 'Heatsheet' exposed to a quiescent fluid

In an early study of heat transfer from inclined plates [19], it is suggested that convection coefficients could be determined from vertical plate correlations, if g is replaced by $g\sin\theta$ in computing the plate Rayleigh number for $0\leq\theta\leq30°$.

3.2.2.5 Working Fluid

The working fluid plays a key role in the thermosyphon heat exchanger. As discussed in Chapter 2, an initial consideration in the identification of a suitable working fluid is the operating temperature range. Since the 'heat sheet panel is designed to operate at sub-atmospheric internal pressures to maintain integrity and prevent separation of the two surfaces, the working fluid must therefore be selected to have its saturation point at sub-atmospheric pressure throughout the design working temperature range. In waste water heat recovery, the 'Heat sheet' is expected to operate with inlet waste water temperatures from as low as 15°C to up to 70°C, therefore the working fluid's saturation temperature at atmospheric pressure should not be less than, say, 60°C to ensure that a sub-atmospheric internal pressure is always maintained, thereby preventing the possibility of the panel bulging and distorting. Some organic and inorganic working fluids are now reviewed for possible use in a 'heat sheet'. Toluene has the strengths of compatibility, stability and acceptable latent heat of vaporization, boiling point and freezing point. Although benzene is more stable than toluene, it has higher toxicity and lower latent heat. Water has high latent heat of vaporization and no contamination problem, but compared with other fluids like toluene, benzene, ethanol, hexane and methanol, its boiling point of 100°C is too high for a waste water system. Acetone has a boiling point of 56.05°C. While its latent heat of vaporization is relatively low when compared to the other fluids considered, acetone is considered non-toxic. It may be easily stored and handled and it is readily available at low cost. It represents a good compromise among the properties desired.

3.2.3 Description of Experiment

A prototype heat exchanger with a single "Heatsheet" panel was designed and made. The system used to perform the experiment was installed in the Thermodynamics Laboratory at University of Canterbury. It consists of separated hot and cold water flow systems, a heat exchanger, and instruments which are discussed in detail below.

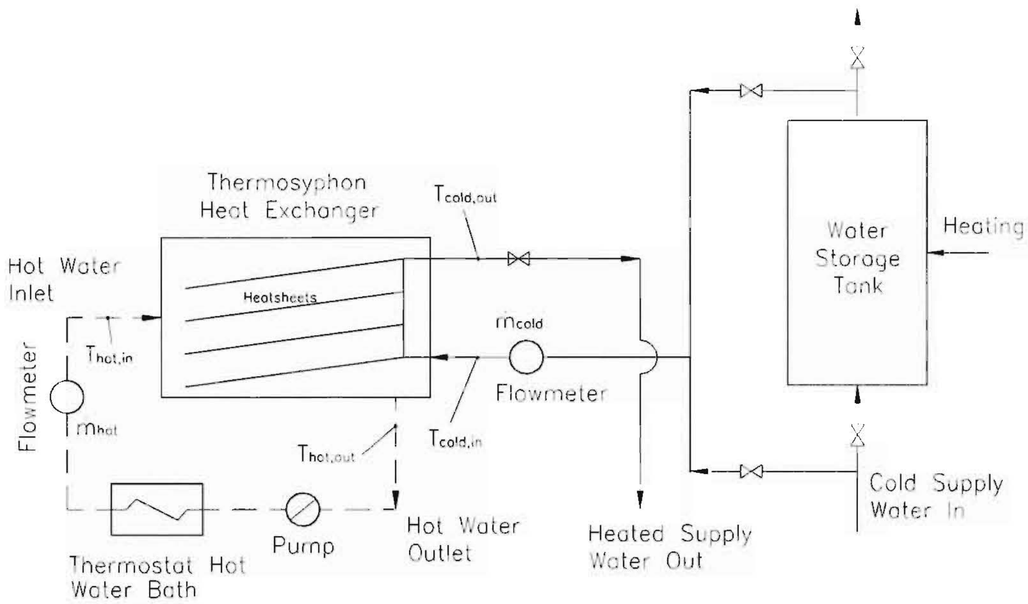


Figure 3.2-2: Prototype heat exchanger system

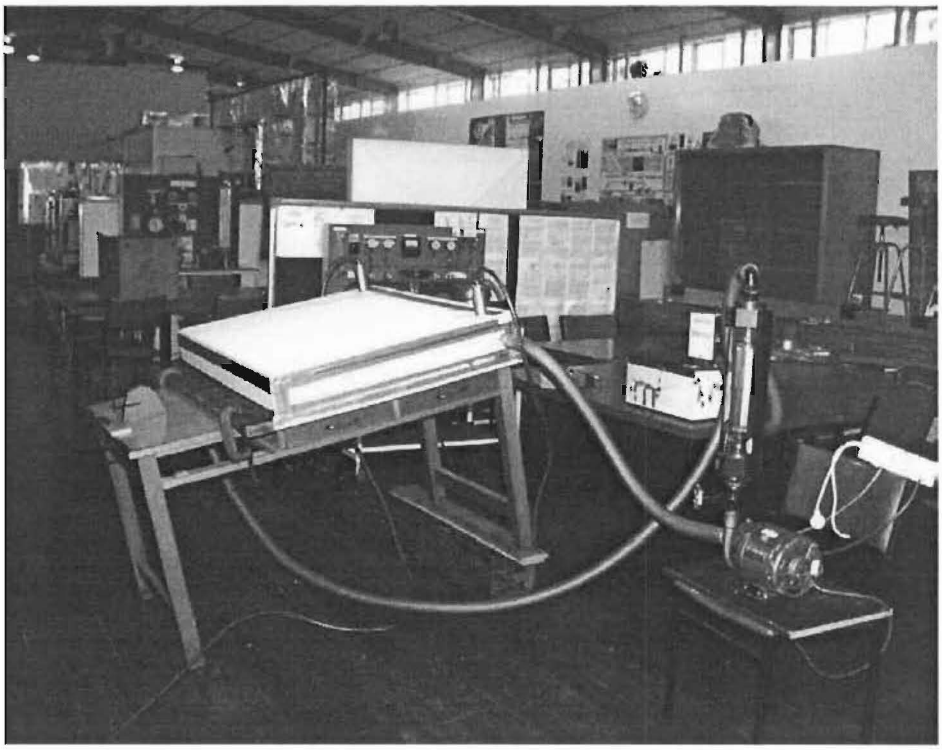


Figure 3.2-3: Photo of a single panel performance set-up

3.2.3.1 Single Panel Heat Exchanger

As introduced in Chapter 2, a thermosyphon 'Heat Sheet' was chosen as the key element of the proposed heat exchanger. The performance characteristics were required to be determined. The "Heatsheet" consisted of two plates seam welded together around the edges to form an envelope that has been evacuated and then load with a small amount of heat transfer fluid and hermetically sealed. Both sheets were punched with a dimple pattern for small pools of the working fluid to accumulate. The material of the sheet plates and supply water tube was stainless steel out of concern for the effects of possible fouling on the heat transfer surface by grease and other contaminants in the waste water.

There was a stainless steel 'Heatsheet' already available in the laboratory for experimental use. It was 915mm long, 735mm wide with the thickness of 2mm (in flat area) and 10mm(in dimpled area). The water supply tube wrapped in the panel was 0.9m long and 19.05 diameter and was made of stainless steel. The panel had an internal volume capacity of about 800ml for vaporization and condensation of working fluid.

The 'Heatsheet' had to be evacuated and the working fluid acetone degassed before it was filled into the panel. The panel was sealed hermetically. Based on the experience of previous research, a volume of 300ml of acetone was used in for the optimum performance of the 'Heatsheet'.

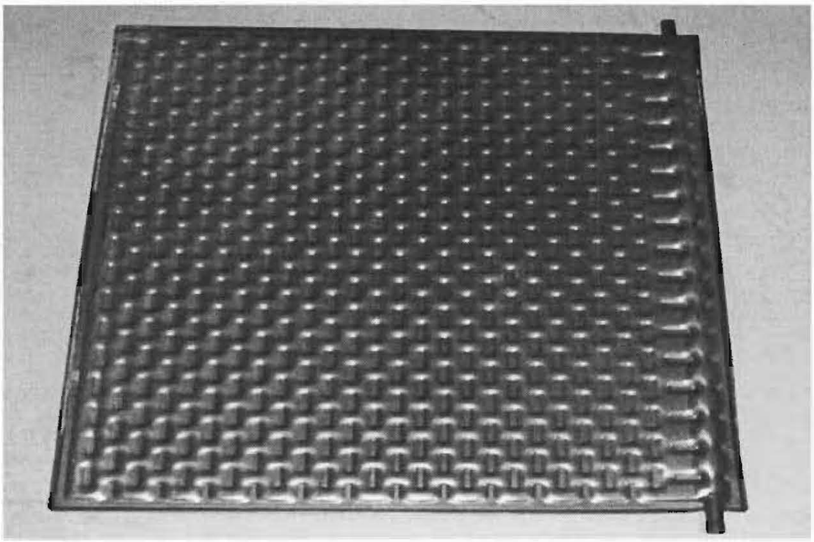


Figure 3.2-4: Dimpled 'Heat Sheet'-size (915mm x 735mm x 4mm)

3.2.3.2 Water Tank and Insulation

The material of the water tank was 1mm thick galvanised steel and the tank size was 970mmL x 800mmW x 100mmH. Two diffuser boxes were designed having holes of varying sizes along the length. They were installed at the inlet and outlet of the hot water side to maintain an even water distribution in the container. The panel was assembled in the middle of the container equidistant from the top and bottom container covers.

The water tank was tightly covered by a layer of 30mm thick polystyrene. All pipes, hot water bath and hot water storage tank were well insulated to reduce heat loss in the system.

3.2.3.3 Water Flow

The single panel heat exchanger performance test included a looped hot water system and an unlooped supply cold water system.

1). The hot water loop consisted of a single panel heat exchanger tank, a water pump, a flowmeter (detailed in Section 3.2.3.4), heating elements, a 500ml volume hot water bath, and valves necessary to conduct the experiments.

The water pump was driven by a 0.33 HP constant-speed motor to provide flows of 0 ~20 l/min.

The hot water bath included a water tank, a 1.5kW heating element, an additional 1.2kW heating element and its power control box, a small internal pump and thermostat that kept heating and circulating hot water in the water bath in order to stabilize hot water output temperatures for performance testing in varied operating conditions.

The temperature was controlled by both the 1.2 kW heater power control box and the hot bath thermostat system.

Complete control over the flow was provided by a ball valve installed upstream of the flowmeter.

2). The cold water system consisted of a hot water storage tank, a regulator and a flow mixer on the tank, a flowmeter and a valve.

A standard hot water storage tank with a capacity of 180 litres was fitted with a 3 kW heating element. It provided heated water to the cold water side for experimental use.

The regulator installed at the water inlet of the tank controlled water pressure. It, and the valve upstream of the flowmeter were used together for adjustment of the flow.

By adjusting the flow mixer, thereby changing the mixing proportions of hot and cold water, the outlet temperature of the mixed water could be adjusted consequently.

3.2.3.4 Instrumentation

Instruments consisted of flow measurement equipment and temperature measurement equipment.

1). Flow measurements

There were three volumetric glass flow meters ('Rotameter" type) used in the experiment.

Volumetric glass flow meters were installed on each hot and cold water pipe line. The standard flow meter A made by Tokyo Keiso Company was installed on the cold water pipe line and was capable of measuring 0.5~4 l/min. For testing at flow rates over 4 l/min, flow meter A was replaced by flow meter B which was able to measure flow rates between 2 and 12 l/min. Flow meter C was installed on the hot water pipeline and was capable of measuring 3~25 l/min.

All flow meters were mounted vertically to ensure measurement accuracy during the experiments. To ensure that the flow meters didn't drift during the experiment, they were also calibrated using the stopwatch and weight method before and after the experiments were under taken. The calibration of flow meter B and C is as shown in Appendix A.

2). Temperature Measurement

Temperature measurements included heat exchanger hot and cold water inlet and outlet temperatures, and an array of thermocouples on the surface of the panel, all connected through a 20-position selector switch to a Yokogawa Model 2455 digital thermometer.

All thermocouples were type K (Chromel-Alumel), suitable for a temperature range of –50~250°C. Twelve thermocouples were to measure the temperatures of both surfaces of the heat exchanger panel to enable the temperature distribution of the panel to be examined. Each measuring end was fixed on the surface of the panel with a plastic coating as insulation.

Four thermocouples were inserted and fixed in the middle of water inlet and outlet tubes rather than mounted in the wall of the outer tubes, because the readings from the former would be more accurate without the influence of tube conduction. They were for measuring water inlet and outlet temperature on both hot and cold water sides. (See Figure 3.2-2)

Prior to their installation, a calibration check on the thermocouples was carried out by submerging the measuring ends together with a PRT sensor into a bath maintained at 0°C and 50°C. Those having a temperature accuracy better than 1% were chosen for experimental use.

3.3 Test Procedure

3.3.1 The Choice of Variables

As discussed in Section 3.2.2, many factors can affect the operation of a single panel heat exchanger heat recovery system. Some of factors are invariant, in a given heat exchanger installation because they are fixed by the materials, dimensions and geometry of the heat exchanger. Others, however, change significantly almost continuously. This series of experiments was aimed at determining which of the numerous factors are likely to contribute significantly to the performance of the single panel heat exchanger. The particular variables investigated in these experiments were:

(a). Inclination Angle of single panel heat exchanger
Experiment Range: -4°, 0°, 5°, 10°, 20°, 30°
This aspect of the experiments was to observe the effect of the inclination angle on the transport behaviour of a single panel heat exchanger in the hope of finding out an optimum inclination angle to improve heat exchanger performance.

(b). Water temperature at hot water side

Experiment Range: 20°C, 30°C, 40°C, 56°C, 63°C

In the case of a commercial dishwasher heat recovery system which will be discussed in the next chapter, the typical waster temperature range was about 15~60 °C. This test was to simulate the possible waste water working temperature that later research.

(c). Water flow rate at hot water side

Experiment Range: 4 l/min~7.5 l/min

The volume of the heat exchanger tank was 8 litres. When the hot water was at a flow rate of 4 l/min, it took 120 seconds to travel from the inlet over the length of the panel to the water outlet of the tank. Therefore we could assume that the heat transfer between the hot water flow and the surface of the panel would be effectively free convection heat transfer in which buoyancy is the principal driving force. In the water heat recovery system, the waster water flow rate entering heat exchanger might vary. This test was to study the effect of a small hot water flow rate change on heat exchanger performance.

(d). Water temperature at cold supply water side

Experiment Range: 15°C, 20°C, 25°C, 30°C, 35°C, 40°C, 45°C

By changing the inlet temperatures of the cold water (in conjunction with the hot water temperatures in (b) above), it was possible to obtain a range of heat transfer rates for the single panel heat exchanger operating with a temperature difference ranging from 5K to 50K between the hot and cold water streams

3.3.2 Start-up Procedure

(a) The inclination angle of the heat exchanger was adjusted to the required operating condition;

(b) The hot water side valves were opened, the heat exchanger tank, pump, water bath and pipes with water were filled up;

(c)Then the cold water side valves and taps were opened, the cold water was run through the pipe line;

(d) The hot water pumps were turned to circulate water in the loop until flow stabilized;

(e) Heating elements were switched on on hot water side, the water in hot water loop was heated to testing temperature, and the flow rate and temperature of cold water were kept constant;

(f) The operating condition was checked and stablized the until meeting experiment requirements;

(g) A set of data was recorded every 10~15 minutes;

(h) After the instruments being adjusted to change one of variables, procedures 6) and 7) were repeated.

3.4 Results and Analysis

3.4.1 Data Analysis on Inclination Angle Test

The first part of the experiment was to test the inclination angle's effect on performance. This part was a critical component in the overall experiment as all the subsequent experiments and analysis were based on the result obtained from this part of the experiments.

The angle was set at -4°", 0°, 5°, 10°, 20° or 30° in order. The cold water was unheated supply water at a temperature of 15~16°C. The hot water flow rate was fixed at 7.4 litre/min. Then the hot supply temperature was set at 20°C, 30°C, 40°C, 56°C to 63°C, and the cold water flow rate was varied between 1 and 4 litre/min at each temperature setting.

Data were recorded and then input to a spreadsheet for calculation and plotting. Equations introduced previously in this chapter were used in order to calculate overall heat transfer coefficients under various conditions.

The effect of inclination angle of the single panel heat exchanger on the overall heat transfer coefficient is demonstrated in Figure 3.4-1. This chart clearly shows the heat transfer coefficient trend versus inclination angle, hot water inlet temperature and cold water flow rate.

Important observations resulting from this series of tests are:

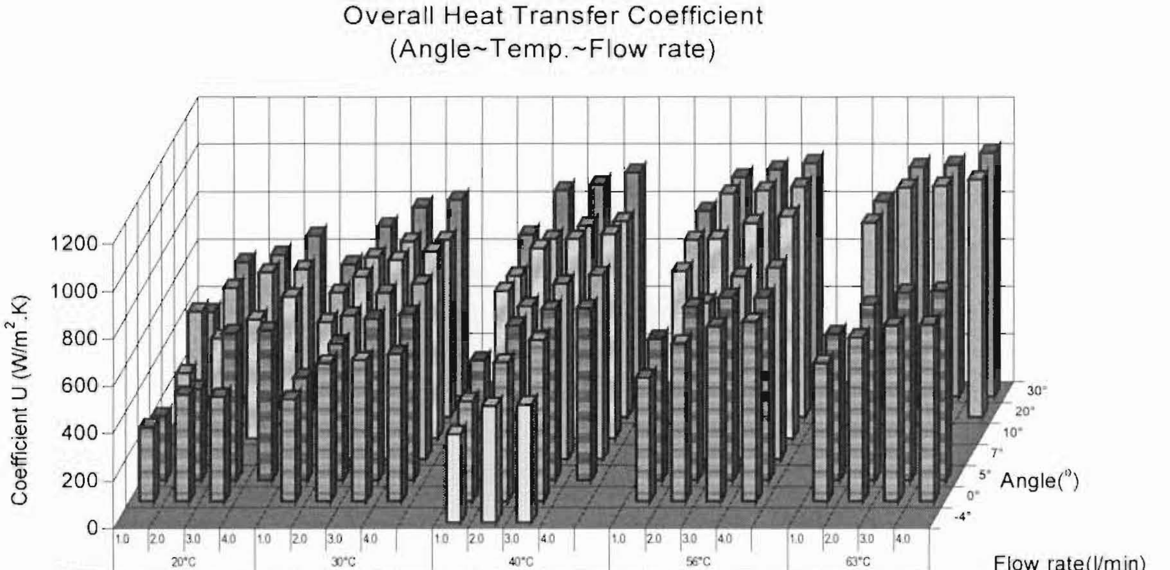
- When the cold water inlet temperature was constant at 15°C, heat transfer coefficients increased with increases in both hot water inlet temperature and cold water flow rate.

- It was found that even for the heat exchanger horizontal (0°) or inclined in the reverse direction (-4°), there was still a significant amount of heat transferred from hot water to cold water.

- Between 7° and 10°' inclination angle a significant rise in the overall coefficient was observed for most of the operating conditions. Beyond 10°, the overall coefficients generally were little different at 20° (in fact in some cases slightly lower) and increased very slightly at 30°.

- The results also indicated that the overall heat transfer rate increased by about 10% when the inclination angle of the heat exchanger increased from 5° to10°. It could be concluded that a peak value of heat transfer coefficient would most likely occur between 7° and 30° with the value difference less than 5%.

- As discussed previously in Section 2.6, when the panel was slightly sloped, more surface of the bottom sheet of the panel was covered by the working fluid. Because the thermal resistance of liquid is much smaller than the thermal resistance of vapor, most of heat would be transferred from the bottom area of the panel. So the overall heat transfer coefficient was mainly based on the bottom side of the panel (this would be discussed in Section 3.4.5). Therefore it was desirable to operate the panel with a slope that was as small as possible. The inclination angle of 10° could be identified as the minimum inclination angle at which good performance was still obtained.



Temp. (Hot side)	20°C				30°C				40°C				56°C				63°C			
Flow rate (l/min)	1.0	2.0	3.0	4.0	1.0	2.0	3.0	4.0	1.0	2.0	3.0	4.0	1.0	2.0	3.0	4.0	1.0	2.0	3.0	4.0
-4°									370	490	495									
0°	310	450	440		430	580	595	620	420	585	680		520	665	735	755	580	690	740	745
5°	270	385	620	630	435	575	680	700	505	650	720	725	595	730	770	770	615	740	790	800
7°						605	700	740		645	740	775		660	770	805				
10°	275	420	500	595	490	680	750	785	620	800	840	860	705	840	905	935				
20°	445	545	610	625	525	675	740	745	595	750	805	825	745	940	955	970	820	965	975	1000
30°	355	565	595	675	555	715	795	825	680	865	890	940	780	920	955	980	820	965	975	1025

Figure 3.4-1: The effect of inclination angle of a single panel heat exchanger

3.4.2 Effect of Temperature Difference Between Hot Water Inlet and Cold Water Inlet- $U \propto \Delta T$

The summary of heat transfer coefficient results is shown in Figure 3.4-2, at the inclination angle of 10 degrees and a hot water flow rate 7.4 l/min. For a test at cold water flow rate of 1 l/min, the heat transfer coefficient went up steadily from 275 $W/m^2.K$ to 705 $W/m^2.K$ at the temperature difference between hot water inlet and cold water inlet increased from 5 to 40 K. When the cold water flowrate was changed from 1 to 4 l/min, the heat transfer coefficient also increased correspondingly. The effect of hot and cold water inlet temperature difference on the heat transfer coefficient could be translated into best fit functions shown below:

U= $0.0001734 * \Delta T^3 - 0.379\Delta T^2 + 29.02\Delta T + 139.3$ at 1 litre/min cold water flowrate;

U=$0.006667 * \Delta T^3 - 0.96\Delta T^2 + 43.03\Delta T + 228$ at 2 litre/min cold water flowrate;

U=$0.01577 * \Delta T^3 - 1.483\Delta T^2 + 49.54\Delta T + 287.4$ at 3 litre/min cold water flowrate;

U=$0.01187 * \Delta T^3 - 1.084\Delta T^2 + 36.82\Delta T + 436.5$ at 4 litre/min cold water flowrate;

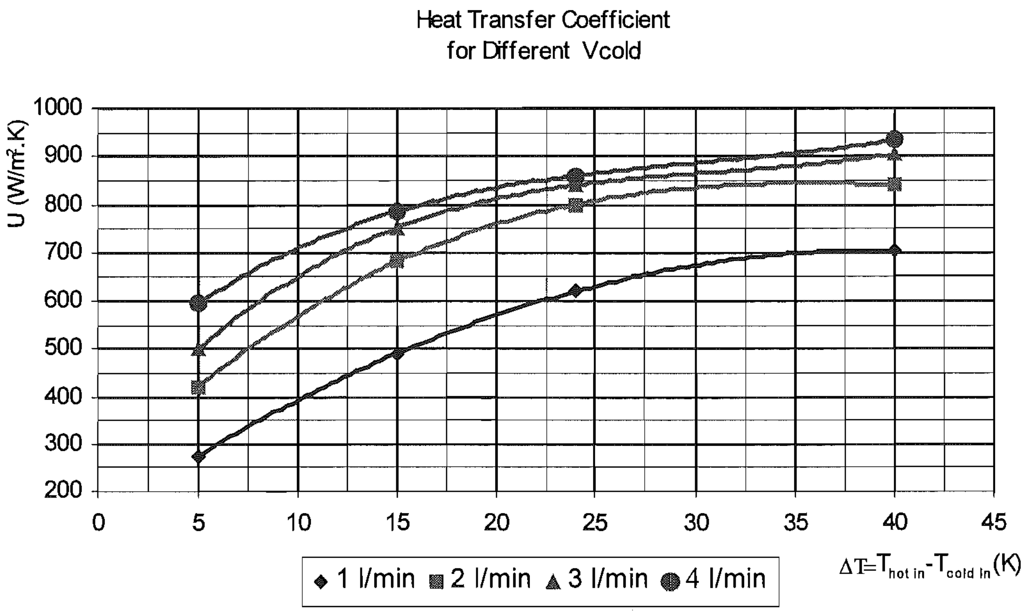


Figure 3.4-2: U versus temperature difference for different cold water flow rates

3.4.3 Hot Water Flow Rate Influence

The hot water flow rate was varied from 4.0litre/min to 7.5litre/min while its temperature remained at 40°C. From Figure 3.4-3, it can seen that when the hot water flow rate is nearly doubled, the overall heat transfer coefficient changes slightly from 970 W/m^2.K to 1035 W/m^2.K for 4l/min, 25°C cold water, and from 970 to 1035 for 8 l/min, 25°C cold water. The influence of the hot water flow rate on the overall heat transfer coefficient is weak due to the large amount of hot water in the heat exchanger and the consequent very low velocity.

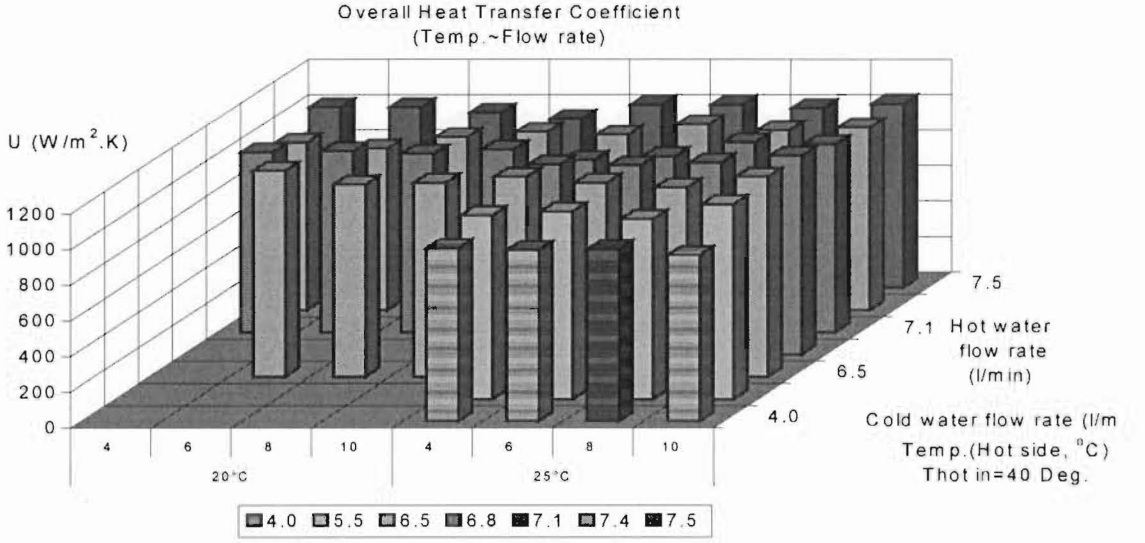


Figure 3.4-3: Hot water flowrate effect on overall heat transfer coefficient

3.4.4 Effectiveness ~NTU Analysis

Heat exchanger effectiveness is a function of Cmin/Cmax and NTU,

where $C\min = \min\{\dot{m}_{cold}c_{p,cold}, \dot{m}_{hot}c_{p,hot}\}$ (Equation 3-6)

$$C\max = \max\{\dot{m}_{cold}c_{p,cold}, \dot{m}_{hot}c_{p,hot}\} \quad \text{(Equation 3-7)}$$

$$NTU = UA / C\min \quad \text{(Equation 3-8)}$$

Figure 3.4-4 shows five effectiveness versus NTU curves that represent Cmin/Cmax 0.27, 0.42, 0.58, 0.85, 0.9, correspondingly. In this chart the curves for Cmin/Cmax 0.42, 0.58, 0.85, 0.9 are short because it would need a large amount of experimental data to complete the profile. It can also been seen these four incomplete curves are reasonably close to the curve which is formed by Cmin/Cmax=0.27(See Figure 3.4-5). Due to limitations of the testing equipment, and also because of time constraints, a functional fit to the most complete set of data (i.e. Cmin/Cmax=0.27) was taken to be an adequate empirical representation of the heat exchanger performance over the full range of operating conditions. It is acknowledged that in using this approximation, effectiveness may be under-estimated when Cmin/Cmax<0.27 and over-estimated when Cmin/Cmax>0.27. However, the range of Cmin/Cmax values collected in dishwasher (discussed later in Chapter 4) was from 0.2 to0.6 and NTU values were between 2 and 6. Therefore most of points in dishwasher data fell close to the curve formed by Cmin/Cmax=0.27. Under the circumstances, it was reasonable to choose Cmin/Cmax=0.27 curve as a general function to represent heat exchanger performance.

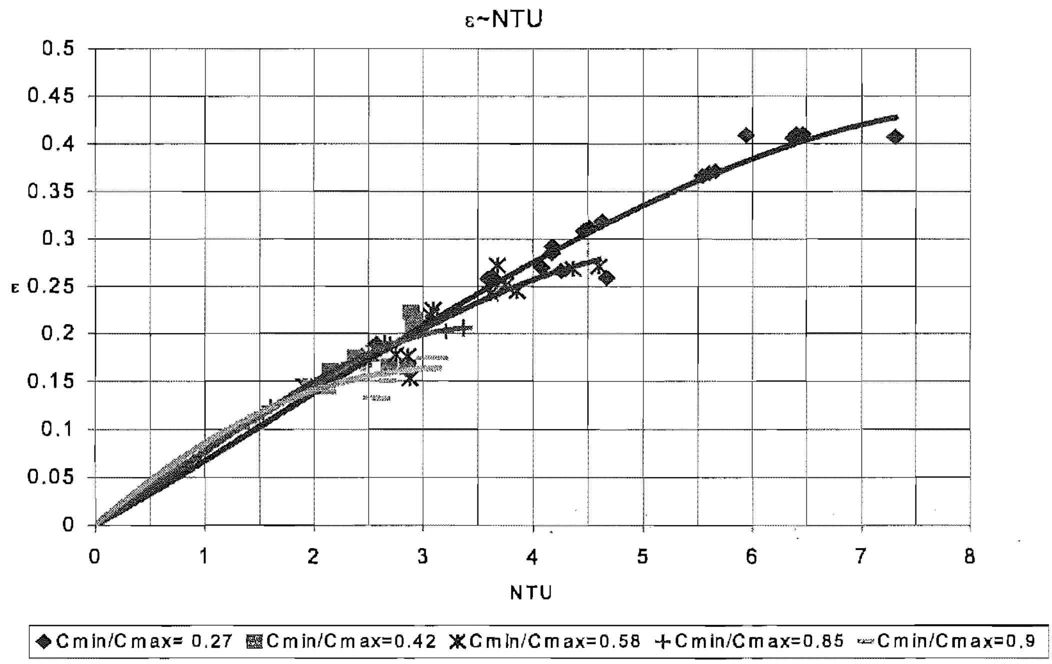


Figure 3.4-4: Effectiveness vs NTU vs Cmin/Cmax

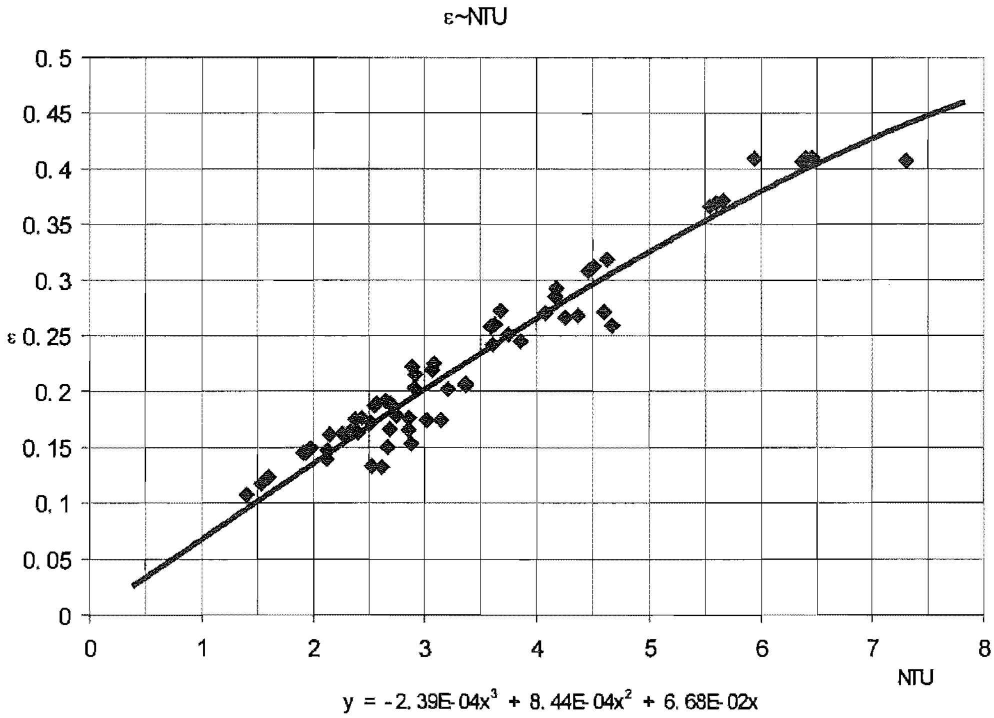


Figure 3.4-5: Effectiveness vs. NTU

3.4.5 Heat Transfer Effect on Both Sides of Single Panel

The experiment was carried out by insulating the top surface of the panel and running the half insulated heat exchanger at the same working condition as that of the original uninsulated heat exchanger.

A 10mm thick polystyrene plate was tightly attached on top surface of the panel. Two layers of thick PVC sheet covered the insulated surface to isolate the top side of panel from heat transfer. The hot water inlet was kept constant at 40°C, 7.4 litre/min while the cold water flow rate from 4 litre/min to 10 litre/min, and the cold water temperature from temperature 20°C to 25°C.

Figure 3.4-6 is a comparison between the overall heat transfer coefficients (U) with top surface insulated and uninsulated in the same working conditions. In the bar chart, the overall heat coefficient when the top surface was insulated and only bottom surface was exposed fluctuated between 960 and 1030 $W/m^2.K$, whereas the overall coefficients values were similar when both surfaces were exposed.

The data therefore could be interpreted as showing that the bottom surface of the panel contributes most of heat transfer while the top surface has little effect on the panel performance. The surface area used in the calculation of U values for top surface insulated and uninsulated in all figures is the sum of the top and bottom surface areas of the panel. Therefore the U values presented in the figures are overall values of the entire panel. Because heat transfers mostly from the bottom surface of the panel and little from the top surface, the U value could be considered as zero for the top surface and twice the plotted U value for the bottom surface of the panel. Hence, even though the top surface of the panel is covered by deposits from the waste water and does not work properly due to the fouling, the performance of the panel would remain the same as heat transfer is based on the lower surface area of the panel. The result is crucial to a multi-panel heat exchanger design in Chapter 5.

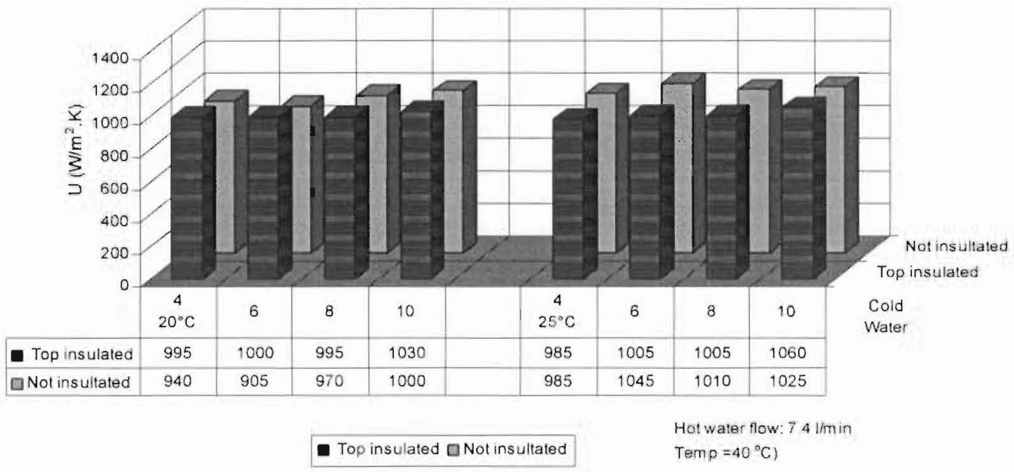


Figure 3.4-6 Heat transfer effect on both sides of panel

3.5 General Discussion

The results of a series of experiments on a single panel have shown that the concept proposed for the waste water heat recovery by using thermosyphon panel could lead to quite effective results.

The inclination angle test has proved that when the single panel heat exchanger was horizontal, it was still able to transfer heat from hot water to cold supply although the amount was little. The inclination angle of 10° could be considered as an optimal working angle for the single panel heat exchanger, as well as the multi-panel heat exchanger.

The major drawback is that the waste hot water outlet temperature was unfavourably high, which indicates there was still a large amount of heat dumped and which could not have been recovered. In other words, the surface area of a single panel heat exchanger was not sufficient as to recover most of the heat from the hot waste water. A multi-panel heat exchanger based on the single panel prototype heat exchanger could provide more surface area to absorb more heat from waste water. The optimal design on the surface area of heat exchanger will be introduced in Chapter 5 & 6.

Convection heat transfer is of course, driven by temperature difference between the two fluids. The thermal performance experiments of the single panel heat exchanger has covered a large range of temperature difference between hot water and cold water, from 5K to 48K. The temperature difference relationship functions calculated from measurements could be generally applied to a single panel as well as multiple-panel heat exchangers.

Although systematic experiments have been done over a range of working conditions, the accurate definition of Effectiveness ~NTU would need a large number of measurements to cover a full range of flowrates, which was limited by both the experimental equipment and time. However, the effectiveness ~ NTU function obtained could be regarded as a general function for all Cmin/Cmax curves for calculation.

Chapter 4: DATA ACQUISITION FROM AN EXISTING DISHWASHER SYSTEM

4.1 Description of the System

4.1.1 Introduction

The main purpose of the project was to introduce a thermosyphon heat exchanger for the development of waste water heat recovery. The principle reason for attempting to recover waste water heat is economic. All waste water heat that is successfully recovered directly substitutes for purchased energy and therefore reduces the consumption of, and the cost of that energy. However, waste water has low temperature and low flow rate. It is uneconomic to install a heat exchanger in a system that doesn't have much energy to recover.

The basic requirements in searching for a suitable heat recovery system as the basis for this research were:

(a) High Waste Temperature.

The temperature of the waste water stream into the thermosyphon heat exchanger has a large effect on heat exchanger performance. The waste water heat temperature must be high enough to serve as a useful heat resource.

(b) Simplicity of System.

The heat recovery system for the research should be simple as a complicated system has lots of factors that may affect the performance and cause some difficulties in energy conservation analysis on the system and the heat exchanger.

Energy required to heat water could be reduced by preheating with the waste water heat from drainlines. Kitchens and laundries at restaurants, and group facilities of hot water flushing at dairy sheds can offer the greatest opportunities for waste water heat recovery since water temperatures are fairly high and heat recovery is predictable.

A search was carried out looking for such commercial kitchen. A dishwashing system in the cafeteria of the Halls of Residence at University of Canterbury came into focus.

The university Halls of Residence provide accommodation for more than 400 students, including three meals a day through the entire study year. The dishwashing system in the kitchen consisted of a Hobart commercial dishwasher and a dish transport rack, hot water supply from a boiler installed in the basement, cold water supply and pre-wash sink, drain lines linked to sewerage(See Figure 4.1-1).

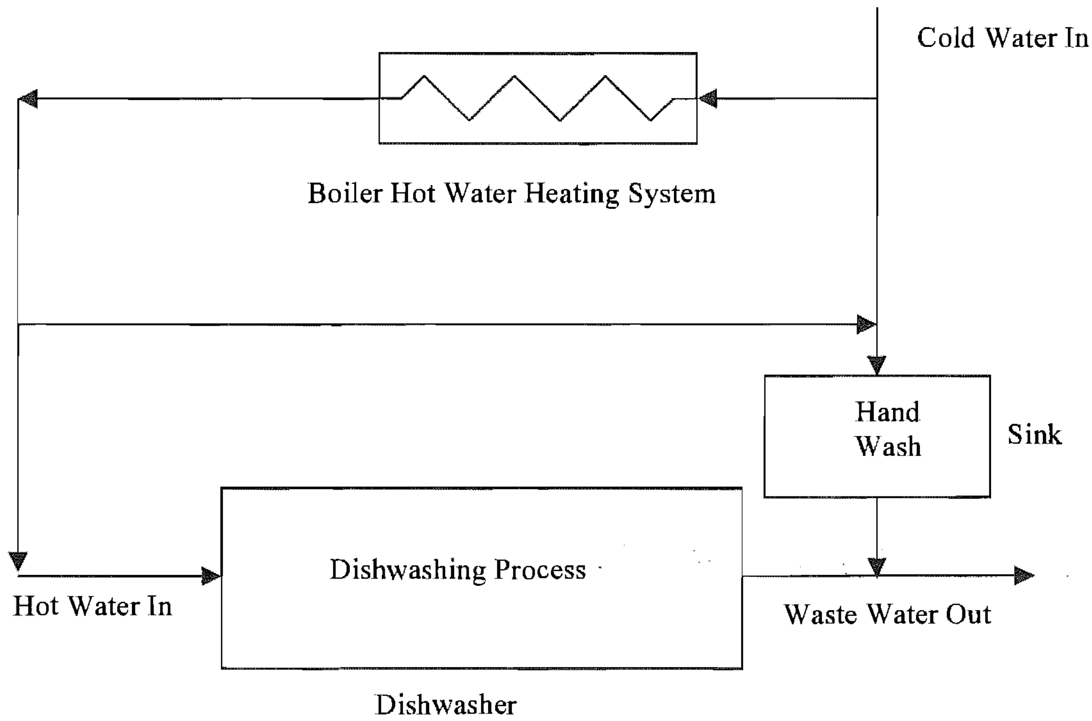


Figure 4.1-1: A Dishwasher water system at Canterbury University Halls of Residence

4.1.2 Description of Dishwasher

The Hobart model C-120R dishwasher (Figure 4.1-2) is a fully automatic, rack type washer that conveys racks of dishes from one end of the machine to the other, exposing the dishes and cutlery to progressive wash/ rinse action. It has a stainless steel tank and chamber with a welded stainless angle frame, stainless steel legs, and stainless steel adjustable feet. A front inspection door provides access to the interior of the wash chambers.

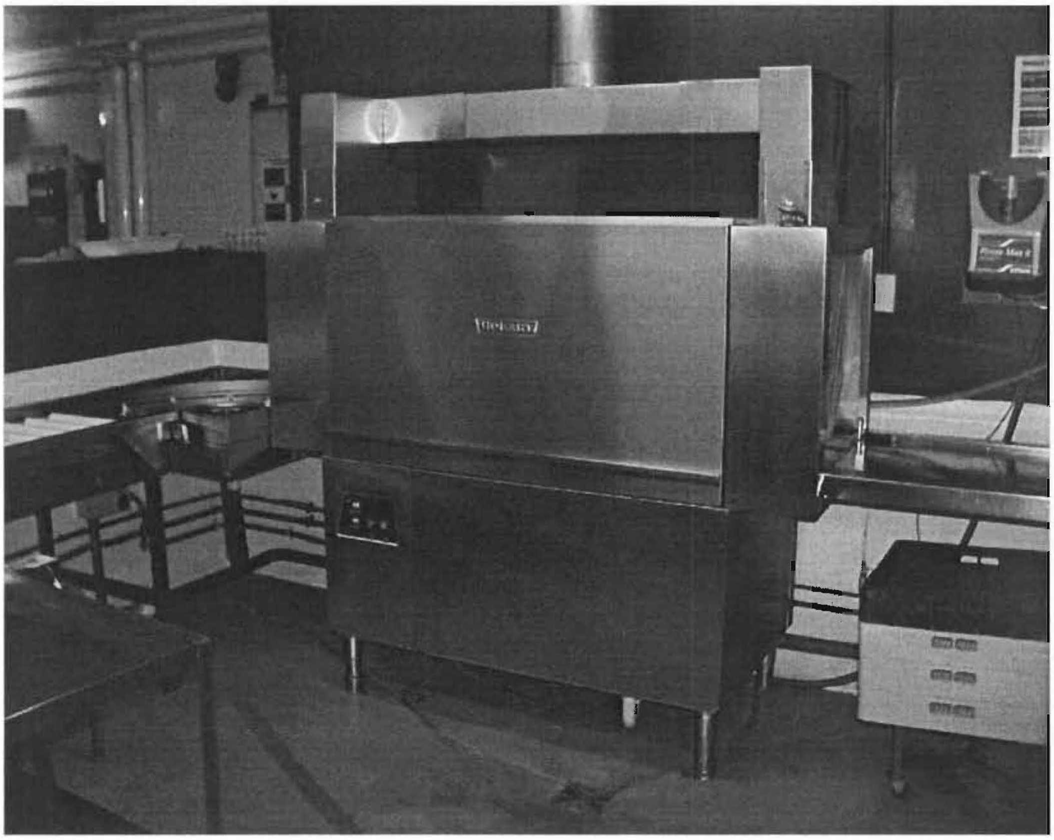


Figure 4.1-2: The Hobart Dishwasher

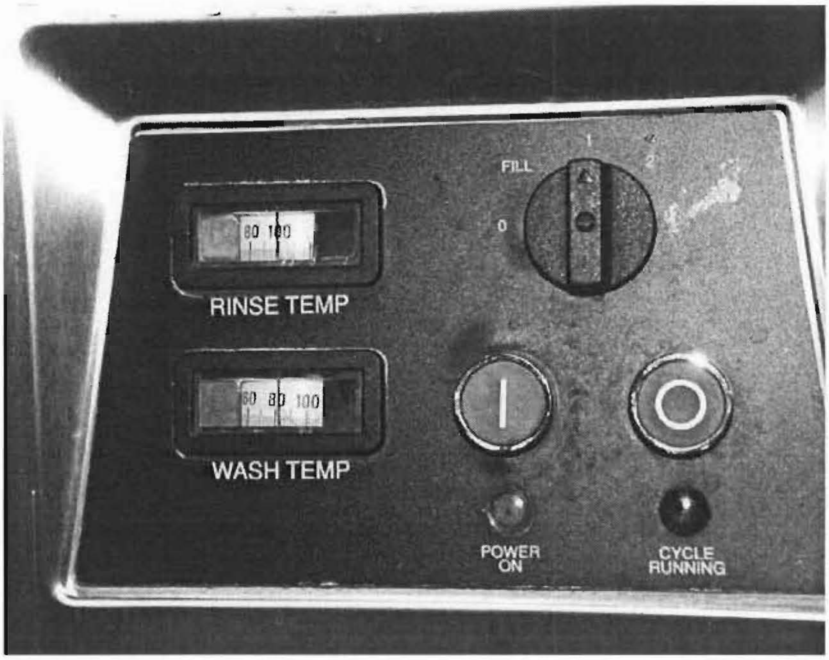


Figure 4.1-3: Controls of dishwasher

The controls are mounted low on the front of the dishwasher (Figure 4.1-3). There are three operational control switches housed in the control box: Power (On-Off); Cycle Running Motor (On-Off); and Process Selection (0, Full, 1, 2, 3). The wash/rinse pump and motor and a pre-wash pump motor are installed inside under the process chamber. Upper and lower pre-wash, wash, and final rinse arms are removable. The 12 litre pre-wash unit is a pre-wash tank fitted with two immersion heaters with a total capacity of 18kW. It has an overflow water pipe installed, so, no fill valve is required.

A stainless steel common drain tube connects the dishwasher and pre-wash drains together, thus requiring only one drain connection at installation.

4.1.3 Process Description

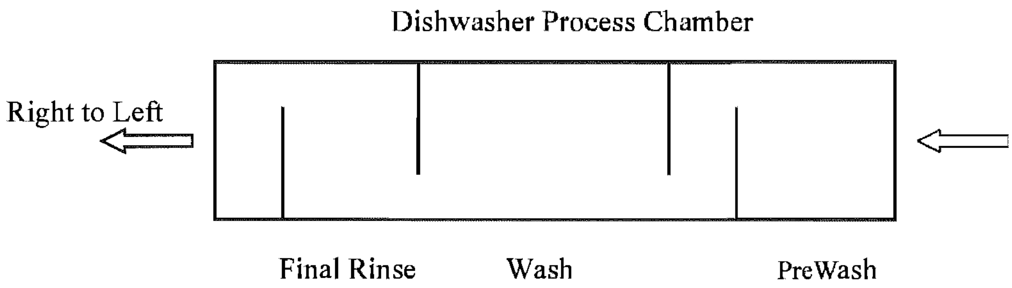


Figure 4.1-4: Schematic of dishwashing process

1). Filling the dishwasher tank

The Power is switched on (Figure 4.1-4), The control switch is then turned to "0". The pumps are left off until the dishwasher pre-wash tank has completely filled.

2).Dishwashing

After the machine has been filled, the pumps are started by pushing the Cycle Running Motor switch on, and the required dishwashing process selected, (Full for full capacity; 1 for slow motion; 2 for faster motion). The dishes are stacked in the racks and the racks are loaded one by one. Each rack moves through prewash, wash, and rinse zones, then out onto the clean dish table. The rinse lever is actuated by the dish rack and automatically shuts off the final rinse water when no rack is in the rinse zone. The dishes are left to drain and air dry before they are removed from rack.

3). Dishwasher cleaning

Water in the machine must be emptied for cleaning at the end of each working shift. The Motor and Power are turned off. After the doors are opened and then the drain is opened by pulling up the drain lever up in the pre-wash tank. After the water has drained, the dishwasher walls and tank are cleaned.

4.2 Installation of Instruments and Data Collection

An experiment was conducted to collect information on water usage of the dishwasher. The various measurements that were made are indicated on the system diagram shown in Figure 4.2-1. This section gives details of the measuring devices.

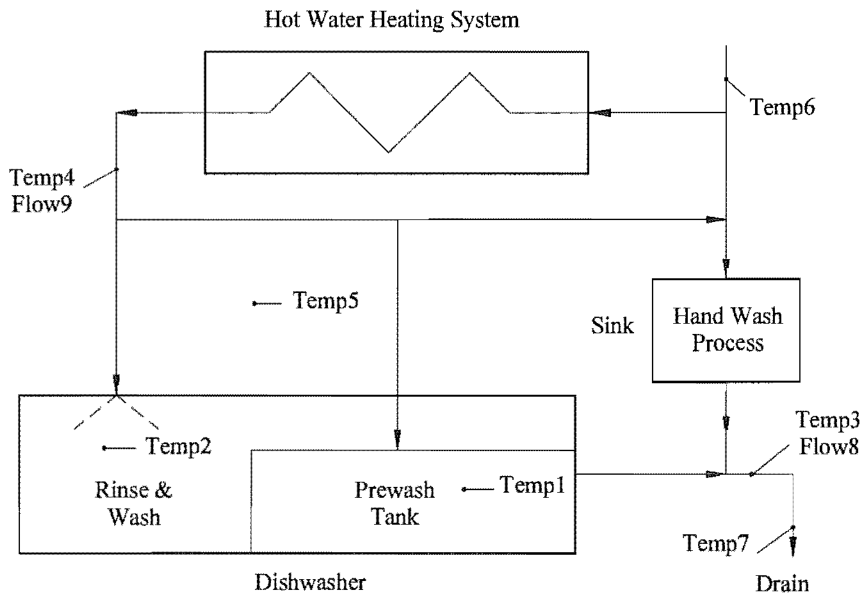


Figure 4.2-1: Dishwasher water system

4.2.1 Data Logging System

The logging system consisted of an Enermet logger and programme for dishwasher hot water flow rate collection from the Enermet flowmeter; a Pico logger and Picolog data logging programme for 7 temperatures and waste water flow rate data collection; and a computer for recording and analysing the data. The Pico data logger was capable in one scan of recording up to 16 channels of analogue data. The resolution could be 13 to 16 bits. Values were signed so, for example, 16 bits gave values in the range from -65,535 to +65,535. The time taken for each measurement increased in proportion to the resolution: 16 bits gave the highest precision results but took the longest time per reading. In this test only nine analogue channels were used and recorded. The time interval of each scan was 30 seconds.

4.2.2 Flowrates

Hot water and cold supply water pipes were disconnected and two magnetic flow metres were installed for obtaining two flow rates (Flow8, 9).

1). Flow8: Waste hot water flow rate in litre/sec. The flowmeter was located on the waste water pipe near the dishwasher drain.
Flow Meter Type: Electromagnetic flow meter with pulsed DC magnetic field excitation in compact and remote converter designs;
Manufacture: Bailey Fisher & GmbH;
Model: CPOA-XE/MAG-XE, Signal Converter 50XE41AAA;
Connection: Dn40 Flanged;

2). Flow 9: Hot water flow rate that enters dishwasher. It was installed on the hot water pipeline before water enters dishwasher.
Flow Meter Type: Pulsed DC magnetic flow meter;
Manufacture: Enermet;
Model: MP 115
Connection: Dn20 Threaded On the hot supply water pipeline.

Both flowmeters were calibrated by comparing measurements using the stopwatch and weight method with measurements using the data logging before installation. As the Enermet flow meter was 0.033 litre/pulse output, when no flow existed the flow meter fluctuated between 0.033 l/min and 0. The accuracy is comparatively low when the flow rate is small and unstable, and is high for large flow rates. The electromagnetic flow meter calibration showed that the flow measuring system has a maximum measurement value deviation of 0.5%. Its calibration curve is show in Appendix A.

4.2.3 Temperatures

Type K thermocouples were used to measure all temperatures in conjunction with the data logging system. Seven temperature points (Temp1 to Temp7 as shown below) were monitored for measuring cold supply water, hot water and waste water temperatures in the pipe work. The following parameters were then output from the data logging programme after each scan for analysis:

1). Day number, day/month/year;

2). Time, hr: min: sec;

3). Temp1: Dishwasher waste water draw-off temperature, °C;
The thermocouple was immersed in the water. The two heating element (totalling 18kW) installed in the tank were to keep the tank at required temperature. The waste water drained through a drain pipe in the tank so it could be considered as the waste hot water draw-off temperature of the dishwasher as well.

4). Temp2: Temperature in rinsing chamber of dishwasher, °C;
The thermocouple was attached on the rinsing pipe under the spray hole. When the dishwasher was not operating, it measured the air temperature in the dishwasher chamber.

5). Temp3: Waste warm water temperature in drain pipe close to dishwasher, °C;
Since the copper drain pipe was not insulated, the temperature at this point in comparison with dishwasher waste outlet temperature, gave an indication of heat loss in this pipe.

6). Temp4: Dishwasher hot water inlet temperature, °C;
A socket was welded on to the insulated pipe for installation of the thermocouple. The thermocouple was inserted into the middle of the water flow in the hot water pipe.

7). Temp5: Ambient temperature, °C;
This was measured in the basement under the kitchen which would be the location if the proposed heat exchanger was to be installed.

8). Temp6: Cold supply water temperature, °C;

This was attached on the water pipeline in the basement and well insulated to improve the precision of the measurement.

9). Temp7: Waste warm water temperature in drainage pipe in the basement, °C;

It was also tightly attached on the pipe and insulated.

4.2.4 Data Collection

Data logging system was installed on the dishwasher pipeline. Water usage was collected continuously 24 hours a day for a week time at a 30 seconds interval from July 18th to July 24th, 2003.

The water usage was evaluated through the following equations:

Volume (at scan i, litre)=Flow rate (at scan i-1, litre/sec)× Time interval (sec/scan)

Overall water usage during a period= $\sum_{i=1}^{n} Volume(i)$

The data were displayed in both graphical and spreadsheet formats. Figures 4.2-3, 4.2-4 and 4.2-5 show the water usage pattern for a week in graphical form.

Table 4.2-1 illustrates the waste warm water discharge volumes of the dishwasher over the week.

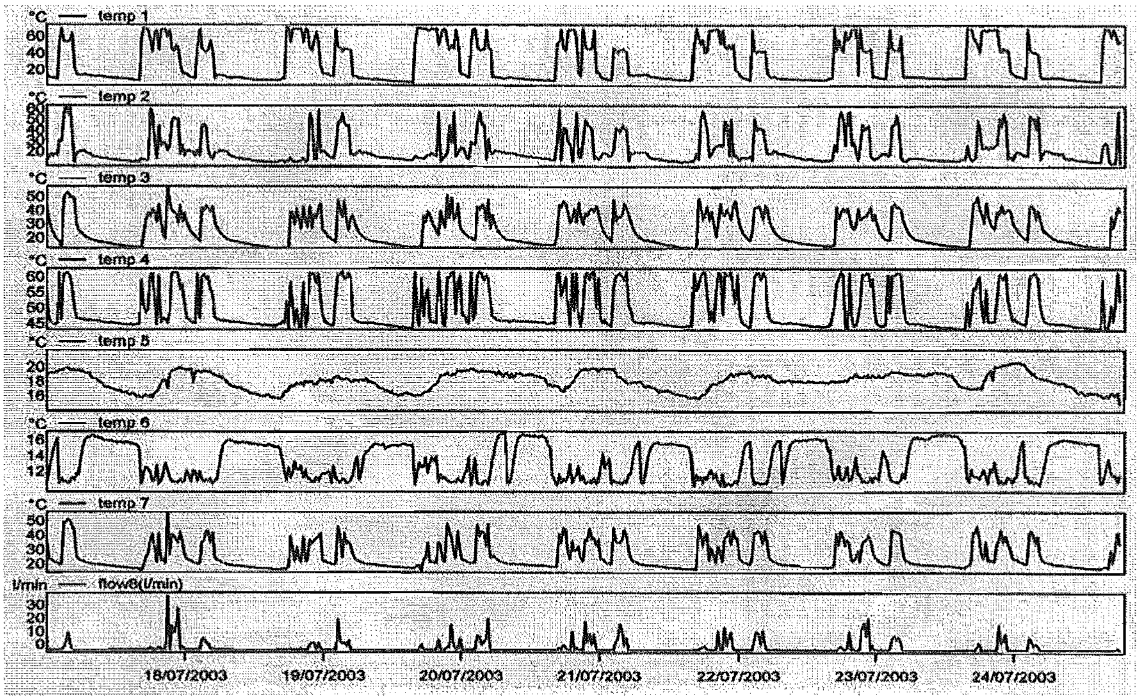


Figure 4.2-2: Dishwasher Water Usage Pattern 1 in a Week

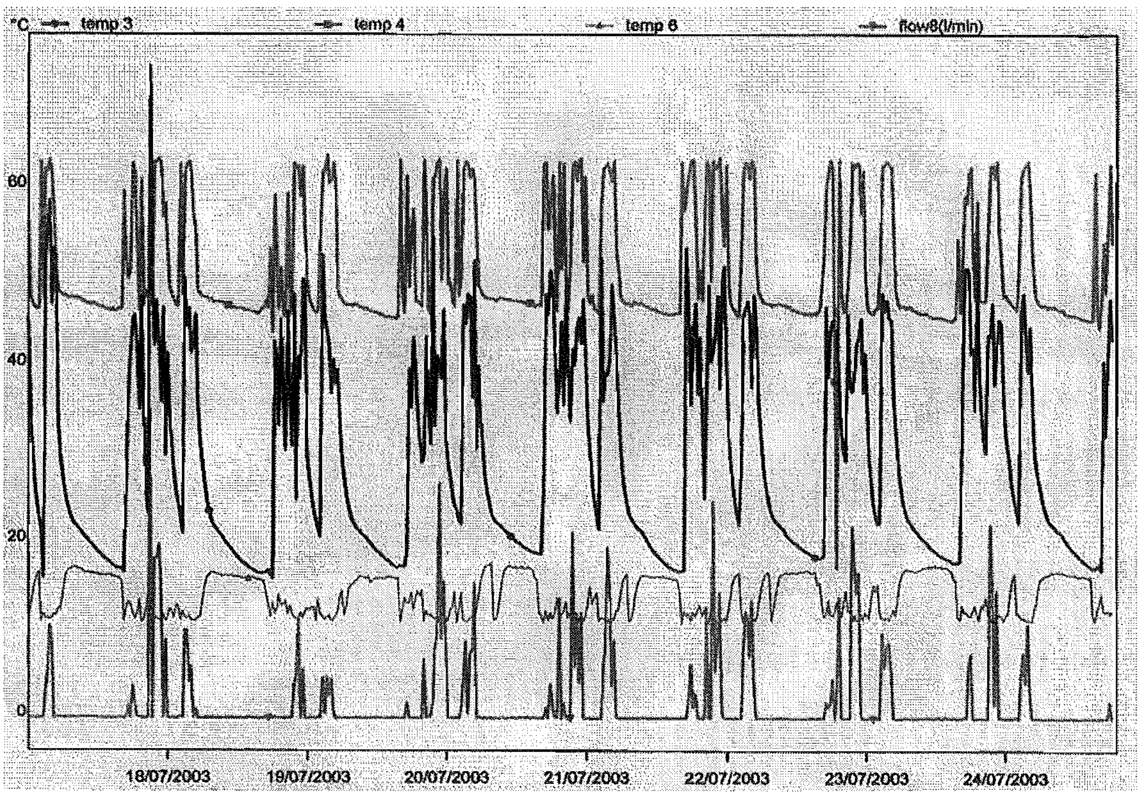


Figure 4.2-3: Dishwasher Water Usage Pattern 2 in a Week

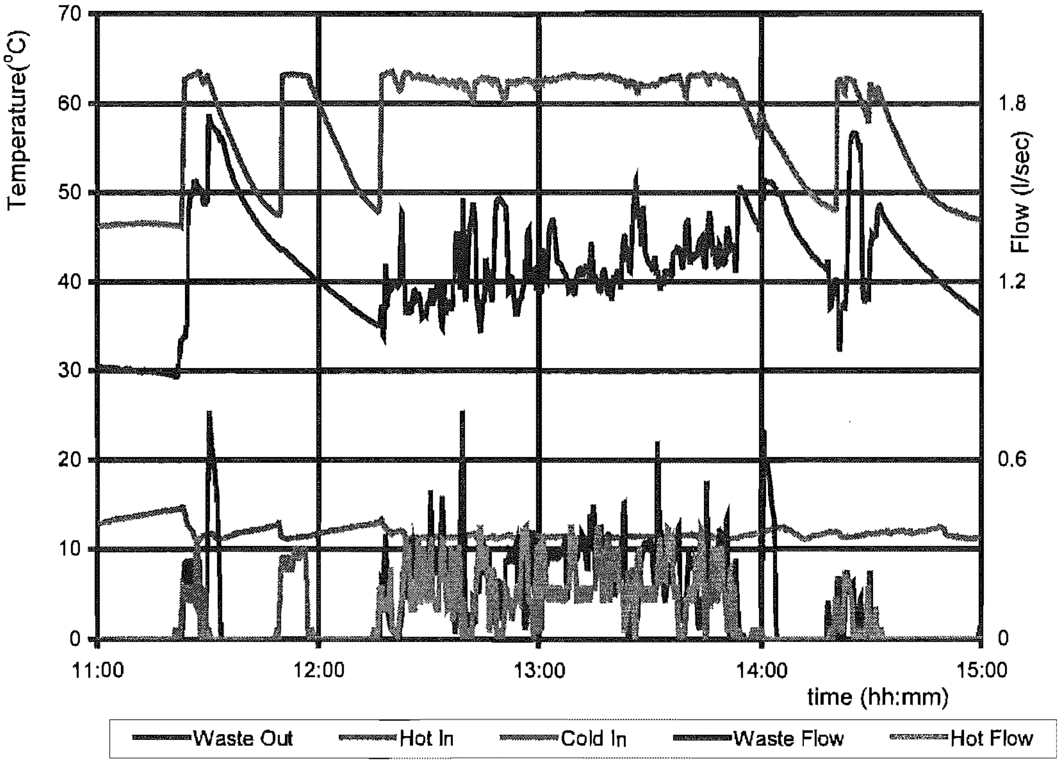


Figure 4.2-4: Dishwasher Water Usage Pattern 3 at a Lunch Time

Table 4.2-1: Waste water discharge volumes of the dishwasher over the week.

Time			Water Discharge (Litre)	
			Each Meal	Total for Day
18-July (Friday)	Breakfast	7:00~10:30	277	2937
	Lunch	11:30~2:30	1659	
	Dinner	5:30~8:00	1001	
19-July (Saturday)	Breakfast	7:00~10:30	79	1611
	Lunch	11:30~2:30	736	
	Dinner	5:30~8:00	796	
20-July (Sunday)	Breakfast	7:00~10:30	28	1679
	Lunch	11:30~2:30	787	
	Dinner	5:30~8:00	865	
21-July (Monday)	Breakfast	7:00~10:30	557	4167
	Lunch	11:30~2:30	1912	
	Dinner	5:30~8:00	1697	
22-July (Tuesday)	Breakfast	7:00~10:30	248	3649
	Lunch	11:30~2:30	1833	
	Dinner	5:30~8:00	1568	
23-July (Wednesday)	Breakfast	7:00~10:30	419	3133
	Lunch	11:30~2:30	1689	
	Dinner	5:30~8:00	1024	
24-July (Thursday)	Breakfast	7:00~10:30	387	2481
	Lunch	11:30~2:30	1406	
	Dinner	5:30~8:00	688	

4.3 Sample Results and Comments

Based on the data recorded between July 18^{th} and July 24^{th}, the following observations can be made:

1) Figures 4.2-3 and 4.2-4 show that temperatures and flow rates collected formed similar curves and the water consumption of the dishwasher for breakfast, lunch and dinner were a similar pattern every day. Figure 4.2-4 is a representative dishwasher water discharge pattern for a lunch. Assuming similarity of the water usage pattern for the dishwater each day would simplify the design of a dishwasher waste water heat recovery system. Thus a day's water usage data could be selected as a representative daily water usage of the dishwasher heat recovery system for the proposed thermosyphon heat exchanger performance analysis.

2) There was no water draw-off during the night period and between two meals. The dishwasher worked from 7 am to 10:30 am for breakfast dishwashing, 11:30am to 2:30pm for lunch washing, and 5:30pm to 8:00pm for dinner washing. We could hence assume there is no energy consumption when there was no water draw-off. Therefore more focus should be put on the water discharge for each of the three meals.

3) Table 4.2-1 shows that the dishwasher had the highest water usage (4167 litres) on Monday (July 21^{st}) while water usage on Saturday and Sunday (July 19, 20) was comparatively low (1611 litres and 1679 litres correspondingly) for this week. It could be explained that some students living in the Halls of Residence were out during weekend and came back to school on Monday. An average water usage of 2450 litres a day could be regarded as a daily water discharge to the proposed heat exchanger for heat recovery performance calculation.

4) Figure 4.2-1 shows the hot water inlet temperature remained stable above 60°C while the cold supply water temperature was constant at about 12°C through the whole meal time. The water draw-off temperature fluctuated between 35°C and 55°C.

When the dishwasher stopped working, the hot water inlet temperature and waste water draw-off temperature sloped down steeply due to large heat loss from the water to the surroundings. As a result, 60°C could be considered as representative temperature value of hot water supply in the dishwasher heat recovery system.

The dishwasher data collected indicated that the dishwasher system had a desirably high waste water discharge temperature. It fell within the working temperature range of the single panel heat exchanger in the component test described in Chapter 3. The performance curves that were obtained could be input into dishwasher data for energy consumption analysis.

The data from the simple dishwasher system would enable heat transfer calculations to be performed for the model simulation of the proposed thermosyphon heat exchanger system being used in conjunction with this dishwasher. However, the data acquired also suggested a smaller than expected amount of waste water discharge. This might bring some difficulties in modelling and calculation in later chapters as the energy saving might not be as large as may have been hoped for.

Chapter 5: MODELLING OF A MULTI-PANEL HEAT EXCHANGER SYSTEM

5.1 Method of Analysis

The component experiment for the single panel discussed in Chapter 3 has shown the single panel heat exchanger has a desirable overall heat transfer rate when its inclination angle is 10°. The bottom surface transfers most of heat while the top surface contributes little on single panel performance.

In a simple waste water heat recovery system proposed in Section 1.4, heat is recovered from warm waste water by a heat exchanger and then is sent to a hot water storage tank for hot water supply. In the dishwasher system (Figure 4.2-1), hot water supplier is a boiler heating system. The warm waste water is directly dumped into the drain with no heat recovery. If we introduce the dishwasher system into the heat recovery system, we could modify the system. In this heat recovery model, the storage tank will replace the boiler to heat water to hot water supply temperature for dishwashing. The waste warm water dumped after the dishwasher is to be delivered to a thermosyphon heat exchanger equipped with a set of panels. Cold water supply recirculates from the bottom of the hot water storage tank to the heat exchanger where it is heated. The heated supply water will then be pumped back to the lower part of hot water storage tank. An energy conservation calculation could be applied to the heat recovery system in which a thermosyphon heat exchanger is installed in comparison with a system without thermosyphon heat exchanger in a search for an optimal thermal design of the heat recovery system. This modelled system is shown in Figure 5.1-1.

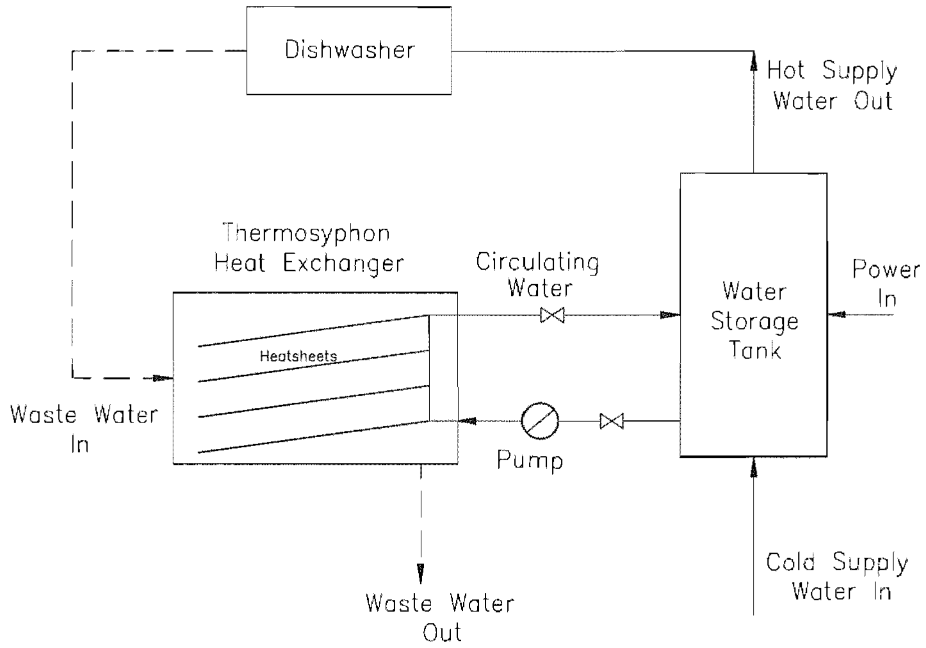


Figure 5.1-1: Schematic of dishwasher waste water heat recovery system

In the single panel experiment results it was noticed that the hot water outlet temperature was only slightly lower than the hot water inlet temperature, even when the cold water flow rate was increased. Furthermore, the temperature difference between the hot water outlet and cold water outlet temperatures was still too large to ignore. These results indicate that there was significant energy left in hot water that had not been recovered. In other words, heat transfer surface area of a single panel was not large enough to take away nearly as much energy as the hot water may be able to provide.

A simulation was carried out for modelling the dishwasher waste water recovery system. Its purpose was to test the theory developed and to consider the optimal configuration based on all the experiments, analysis and results of which were stated in Chapter 3 & 4. A computer programme was written for system modelling. It includes a main programme; a multi-panel heat exchanger programme; and a hot water storage tank programme.

The multi-panel heat exchanger programme is developed for a multi-panel heat exchanger simulation. It is to calculate the heat transfer on each panel and then the entire heat exchanger in order to find out the optimal performance for a multi-panel heat exchanger.

The hot water storage tank programme is to simulate the mixing process of cold water in the lower part of the tank and stratification process of heated water in the upper part of the tank. The behaviour within the tank has a clear influence on the dishwashing and the temperature of the water pumped to the heat exchanger and hence on the heat exchanger performance and the system overall. Within the tank itself, it also affects the circumstances under which the thermostat will turn the tank element on and off, and the delivery temperature to the dishwasher as well, so the times at which supplementary heating will be required will also be affected. Therefore the modelling of the hot water tank is crucial to the simulation of the entire system.

The main programme is to link all other programmes together for performance modelling of the dishwasher system with an exchanger employed in comparison with the same system but without the heat exchanger.

The computer software package MATLAB was considered to be the most suitable programming tool for simulation.

5.2 Simulation of a Multi-Panel Heat Exchange

A direct way for achieving more surface area is to add more thermosyphon panels into the heat exchanger (See Figure 5.2-1). The heat exchanger contains a set of thermosyphon panels that are slightly downward sloped. Cold water pipes in the condenser region of the thermosyphon panels are linked together in series. Therefore the cold water inlet temperature of an upper panel is the outlet temperature of the panel below it. The waste warm water enters through a diffuser into the heat exchanger to allow water to thermally stratify. The slightly sloped installed panels could not only perform with thermal diode characteristic described in Section 2.6, but also would cause minimal disturbance of the waste water stratification. The stratified waste water passes through each channel formed by two close panels. The temperature of waste water that enters the bottom channel is slightly lower than that in the channels between the higher panels. However, the lowest panel has the coldest supply water inlet temperature and thus the largest temperature difference between the hot and the cold compared to higher panels. As a result, the lowest panel should transfer more heat to the cold water which would bring a higher temperature increase than the other panels could achieve. The cold water temperature keeps climbing up after passing through several thermosyphon panels. Meanwhile the hot water outlet temperature would be brought further down due to sufficiently large heat transfer surface area of panels and more heat being taken away. The cooled waste water is drawn off through the drain pipe installed at the bottom of the heat exchanger. The heat exchanger tank is filled with hot waste water during the day and will empty cooled water at night when there is insufficient waste heat to recover. By installing a certain number of thermosyphon panels, more heat transfer surface area is created and hence more heat could be recovered. The initial number of panels set in the programme is four.

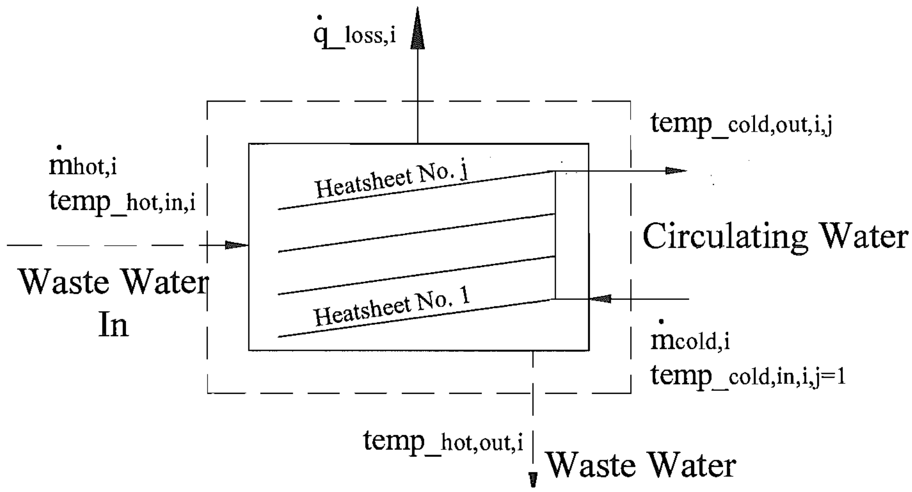


Figure 5.2-1: Schematic of multi-panel heat exchanger process

5.2.1 Heat Transfer Equations for Programme

When the temperature difference between the waste water and the cold water in the heat exchanger is larger than 5K, the waste heat energy is considered worth recovering when this criterion is satisfied, in the simulation. A recirculating pump installed on the cold water pipe line is "switched on" automatically, pumping cold water from the bottom chamber of the hot water cylinder to the heat exchanger. The heat exchanger starts working.

1) An energy balance applied on panel number j of the heat exchanger at time i is:

$$\dot{Q}_{i,j} = \varepsilon_{i,j} C_{min,i,j} (T_{hot,in,i,j} - T_{cold,in,i,j}) \qquad \text{(Equation 5-1)}$$

where,

$$\varepsilon_{i,j} = -2.39 \times 10^{-4} NTU_{i,j}^3 + 8.44 \times 10^{-4} NTU_{i,j}^2 + 6.68 \times 10^{-2} NTU_{i,j}$$

(Section 3.4.4)

$$NTU_{i,j} = U_{i,j} A / C_{min,i,j} \qquad \text{(Equation 5-2)}$$

$$C_{min,i,j} = \min\{\dot{m}_{cold,i} c_{p,cold,i,j}, \dot{m}_{hot,i} c_{p,hot,i,j}\} \qquad \text{(Equation 5-3}$$

$$C_{max,i} = \max\{\dot{m}_{cold,i} c_{p,cold,i}, \dot{m}_{hot,i} c_{p,hot,i}\} \qquad \text{(Equation 5-4)}$$

$U_{i,j}$ could be obtained from Figure 3.4-2.

2) Allowing for the fact that the circulating pump will require an energy input, the energy saved on panel number j of the heat exchanger during time i is:

$$E_{i,j} = (\dot{Q}_{i,j} - \dot{Q}_{pump,i}) t / 3600 \qquad \text{(Equation 5-5)}$$

where,

t is the time interval of data input which is 30 seconds in this programme.

3) Cold water outlet temperature of the panel j at time i is:

$$T_{cold,out,i,j} = \dot{Q}_{i,j} / C_{cold,i,j} + T_{cold,in,i,j} \qquad \text{(Equation 5-6)}$$

$$C_{cold,i,j} = \dot{m}_{cold,i} c_{p,cold,i,j} \qquad \text{(Equation 5-7)}$$

4) Waste water outlet temperature of the panel j at time i is:

$$T_{hot,out,i,j} = T_{hot,in,i,j} - t(\dot{Q}_{i,j} - \dot{Q}_{loss,i,j})/(\rho_{i,j} c_{p\,i,j} V_j) \qquad \text{(Equation 5-8)}$$

V_j is the waste water volume for heat transfer on the panel j;

$$\dot{Q}_{loss,i,j} = \frac{kA_{tank}}{\Delta x_{poly}}(T_{hot,avg,i,j} - T_{air,i}) \qquad \text{(Equation 5-9)}$$

where,

k=0.04 W/m.K is thermal conductivity for foamed polystyrene,

Δx_{poly}, is the thickness of the polystyrene, m.

The heat exchanger tank is assumed to be thin steel plate and covered by 35mm thick polystyrene. The convective resistances are negligible in comparison with the conductive resistance of the polystyrene.

5) The energy that the multi-panel heat exchanger will save in a day:

$$E = \sum_{i=1}^{n} \sum_{j=1}^{k} E_{i,j} \qquad \text{(Equation 5-10)}$$

where,

k, the number of thermosyphon panels installed in heat exchanger; (4 initially);

n, the number of time intervals in 24 hours a day, which is 2880 in programme.

5.2.2 Main Flow Chart for Multi-Panel Heat Exchanger & System

Figure 5.2-2 is a flow chart representing the heat transfer process of a multi-panel exchanger. It is intended to be a schematic way of describing the underlying algorithms

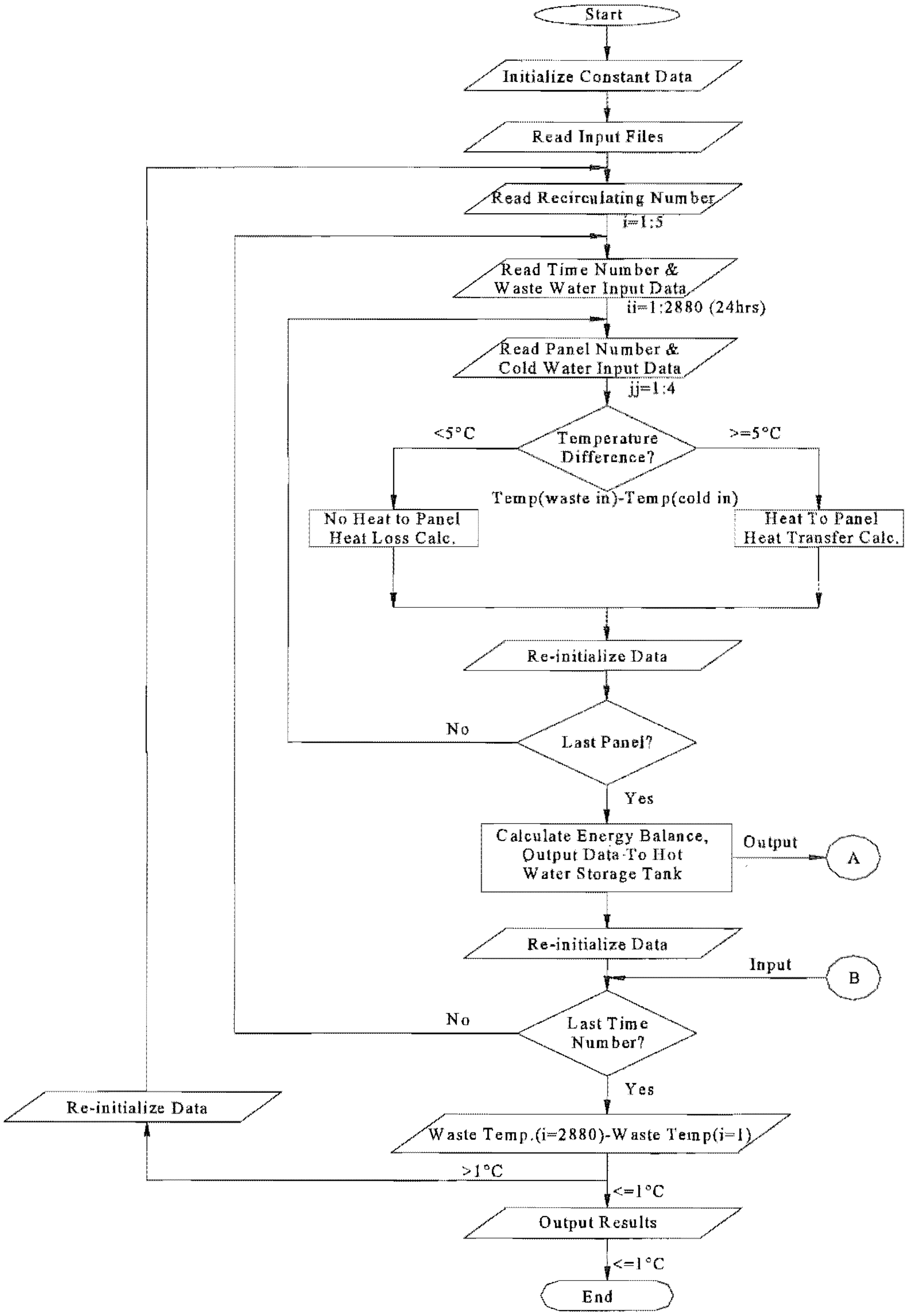


Figure 5.2-2: Main flow chart of multi-panel heat exchanger & system programme

5.2.3 Programme Listing

The programmes written in MATLAB to implement the steps set out in the programme flow chart have been included as Appendix B.

5.3 Simulation of Hot Water Storage Heater

The objective of the simulation of hot water storage heater is to estimate the amount of thermal energy that can be stored in the tank and determine the optimal volume and heating capacities so that the hot water supply can meet the hot water demand of the dishwasher system.

A standard electric storage water heater was employed in the dishwasher waste water heat recovery system in place of the existing boiler heating unit for simplification of the system. It consisted of three chambers (from the bottom to the top): mixing chamber, heating chamber and stratification chamber (See Figure 5.3-1).

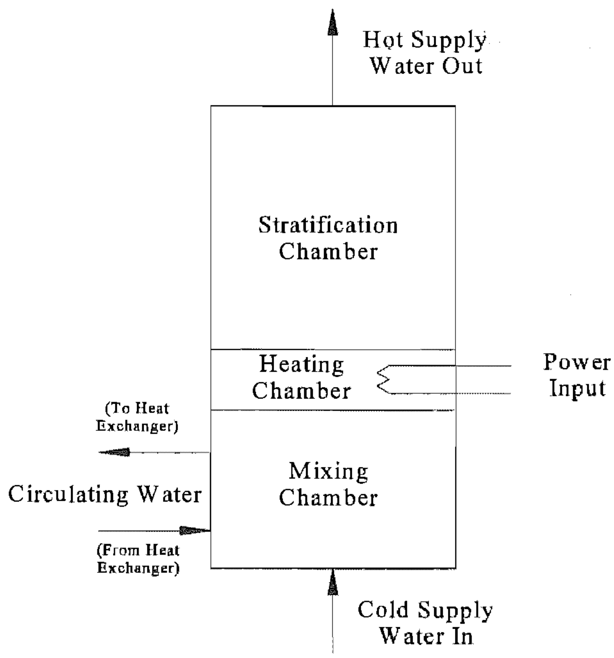


Figure 5.3-1: Schematic of hot water storage heater

In the water storage heater, the water is pumped from the lower portion of the mixing chamber to the multi-panel heat exchanger where it is heated and is returned to the upper portion of the mixing chamber. In the mixing chamber, the cold supply water and heated circulating water are considered perfectly mixed. The electric powered circulation pump is controlled by sensors installed in the heat exchanger tank and the mixing chamber of the water storage heater. If the mixed supply water temperature is too close to the hot waste water temperature, the temperature increase on the cold water side of the heat exchanger might be as small as 0.2 K, which would be hard to measure due to the inaccuracy caused by temperature fluctuation and cause frequent on-off switching of the circulating pump. Therefore, an initial temperature differential setting of 5 K between the hot waste water and the mixed supply water is made, assuming the heat exchanger works only when the water temperature difference between the hot waste water and the mixed supply water is larger than an 5K. Although the chosen temperature differential of 5K would have the consequence of losing some potential energy savings it is a compromise between that and over-frequent cycling of the pump. It is, in fact, typical of the temperature dead-band that is used in thermostatic control of hot water system devices.

When the hot water is in use, the mixed water flows up to the heating chamber where one or more electric resistance elements are immersed. A thermostat in the stratification chamber controls the heating process in this chamber. Heating elements are turned on when the hot water supply temperature is below a programme lower set point (58°C) or heavy draw-offs of hot water are made, and turned off when the water temperature reaches an upper set point of 60°C. Thus the thermostat controlling the heating element has a "dead band" of (60-58) or 2K.

The hot water is stored in the stratification chamber after being heated. The water in this chamber is fully stratified due to buoyancy forces. A temperature 'step' occurs between the upper and bottom part of the chamber. Some researchers [20, 21] suggested a one-dimensional model in which the temperature distribution in the stratification chamber is dependent on the height of the cylinder. Another [22] suggested conditionally dividing the volume of the stratification chamber into several sections according to the height.

However, a report presented by Argonne National Laboratory [23] showed that temperature distribution curve in the stratification chamber was close to being one-dimensional. The water temperature is a function of elevation and time, only. However, when there was high hot water flow rate, the contents of the water tank mixed, and stratification was destroyed, and the water profile changed from stratified to fully mixed.

Based on previous research and also for simplification of the simulation, the hot water temperature in this chamber is assumed uniformly distributed initially when the temperature remains at about 60°C and the hot water storage heater is on stand-by. When there is heavy water draw-off, colder water is pushed up into the stratification chamber, and the programme automatically divides the chamber into two sections. Water in the top section still remains uniform while the water in the bottom section is heated by heating elements until the water temperature of the bottom section reaches the water temperature of the top section, then the temperature in the chamber turns back to being uniform again.

5.3.1 Energy Balance in the Modelling of the Hot Water Storage Heater

Energy Conservation:

$$\Delta E_{store,i} = E_{in,i} - E_{out,i} \qquad \text{(Equation 5-11)}$$

1) Mixing Chamber (Figure 5.3-2)

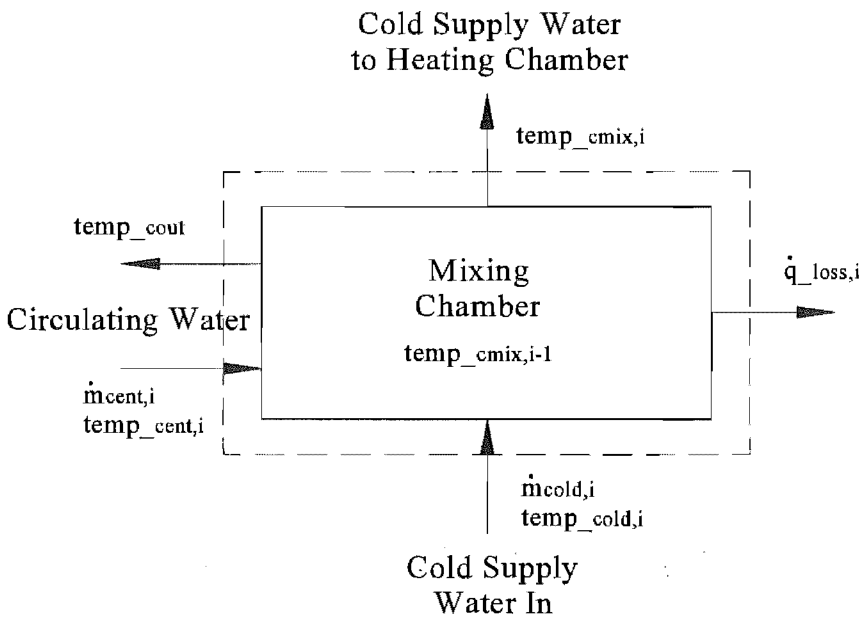


Figure 5.3-2: Water mixing process

$$\Delta E_{store,i} = \rho_{mix,i} V_{mix} C_{p\,mix,i} (T_{mix,i} - T_{mix,i-1}) \qquad \text{(Equation 5-12)}$$

$$E_{in,i} = (\dot{m}_{cent,i} C_{p\,cout,i} T_{cout,i} + \dot{m}_{cold,i} C_{p\,cold,i} T_{cin,i})t \qquad \text{(Equation 5-13)}$$

$$E_{out,i} = (\dot{m}_{cent,i} C_{p\,cent,i} T_{cent,i} + \dot{m}_{cold,i} C_{p\,cmix,i} T_{cmix,i} + \dot{q}_{loss,i})t \qquad \text{(Equation 5-14)}$$

Initial settings:

$T_{cent,i=1} = T_{cold,i=1}$;

$T_{mix,i=1} = 0$;

$T_{cent,i} = T_{cold,in,i}$;

$T_{cout,i} = T_{cold,out,i}$;

where,

$T_{cent,i}$, the temperature of circulating water out of the mixing chamber equals to $T_{cold,in,i}$ (Section 5.2.1), the temperature of water entering heat exchanger at time i;

$T_{cout,i}$, the temperature of circulating water entering the mixing chamber equals to $T_{cold,out,i}$ (Section 5.2.1),the temperature of water out of heat exchanger at time i.

Heat loss from circulating pipes is considered negligible.

2) Heating Chamber and Stratification Chamber (Figure 5.3-3)

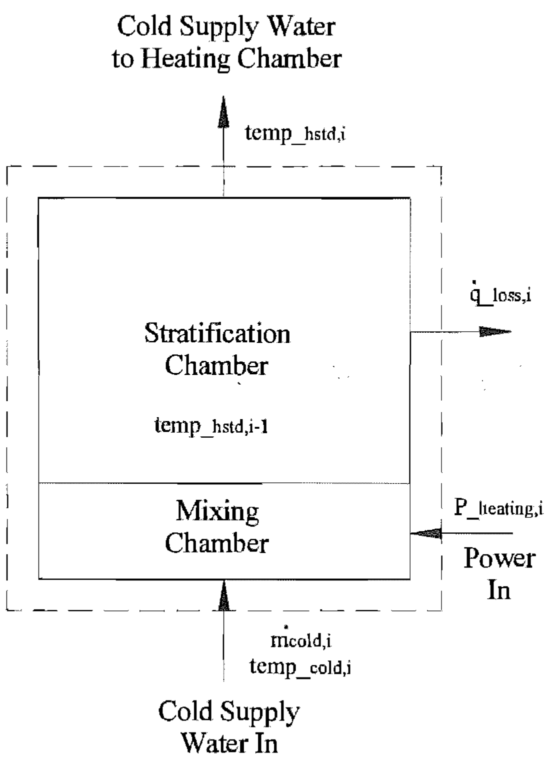


Figure 5.3-3: Heating and thermostat process

$$\Delta E_{store,i} = \rho_{hstd,i} V_{heating} C_{p\,hstd,i} (T_{hstd,i} - T_{hstd,i-1})$$ (Equation 5-15)

$$E_{in,i} = (P_{heating} + \dot{m}_{cold,i} C_{p\,cmix,i} T_{cmix,i})t$$ (Equation 5-16)

$$E_{out,i} = (\dot{m}_{cold,i} C_{p\,hout,i} T_{hout,i} + \dot{q}_{loss,i})t$$ (Equation 5-17)

Initial settings: $T_{hstd,i=1} = T_{hout,i=1} = 60°C$;

$P_{heating,i=1} = 0$;

3) Heat Loss

The cylinder of a hot water storage heater is required to be well insulated. According to the New Zealand hot water storage heater standard, the maximum permitted 24h heat loss shall be determined by the following formulae and rounded to the nearest 0.1 kWh above [24]:

Capacity (L)	Heat Loss (kWh)
≤ 90	0.0084L+0.40
≥ 90	0.0048L+0.72

5.3.2 Sub-Flow Chart for Hot Water Storage Heater

A flow chart for the hot water storage heater was used to describe the flow of data through the MATLAB program, showing the exact sequence of operations performed by the sub-programme on the analysis of heat transfer process.

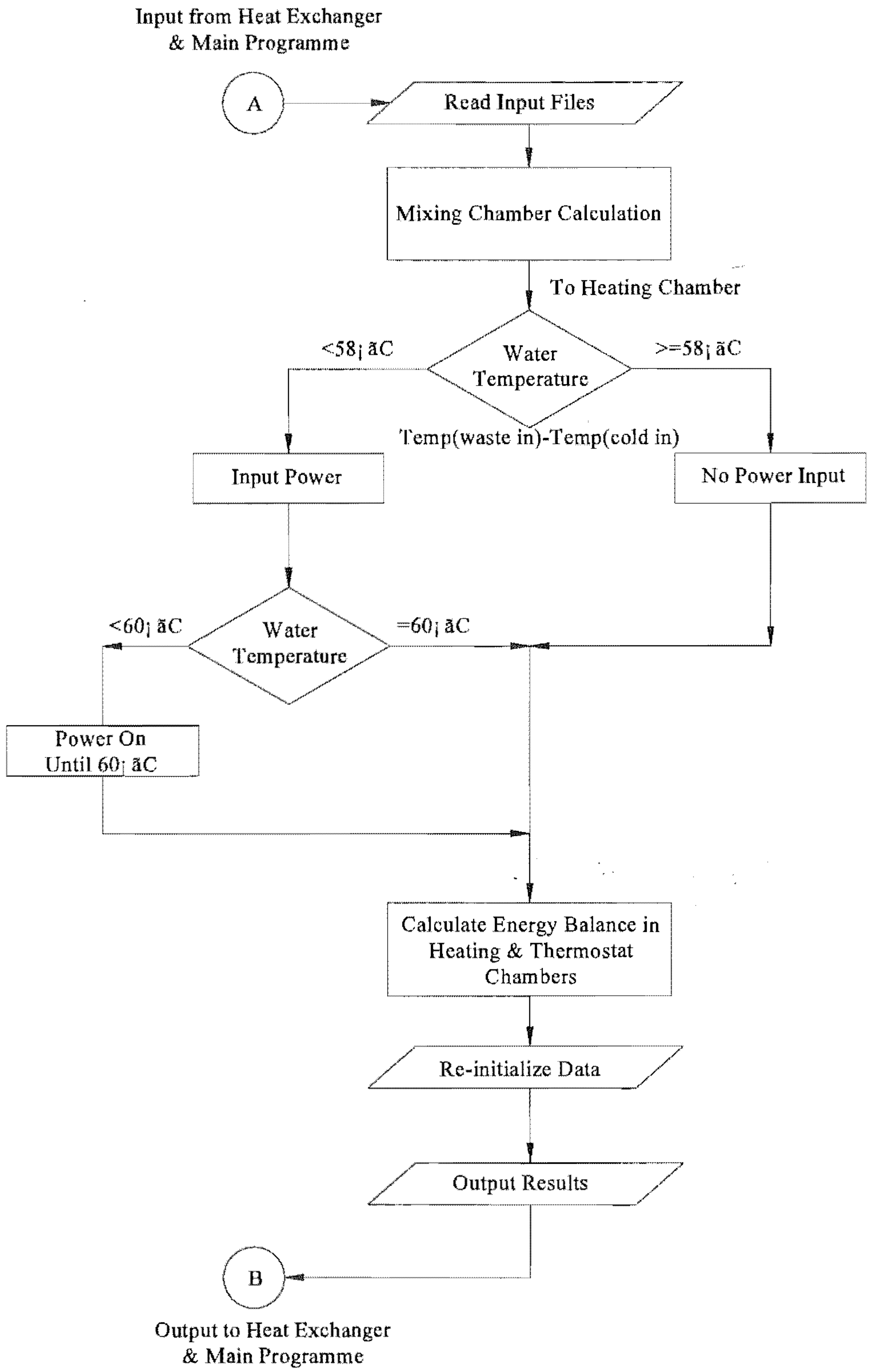


Figure 5.3-4: Flow Chart of Hot Water Storage Heater Programme

(In conjunction with Figure 5.2-2)

5.3.3 Sub-Programme Listing

The sub-programme of hot water storage heater written in MATLAB has been included in Appendix B.

Chapter 6: DISCUSSION OF SIMULATION RESULTS

6.1 Model Comparison with Component Experiment Results

A comparison was made between the results obtained from the heat exchanger model and the results from the single-panel component experiment. Some of the original experimental data were chosen to put into the simplified heat exchanger computer programme for calculating single panel performance in comparison with the experimental readings. Some simulated results and the experiments results are tabulated in Table 6.1-1:

Table 6.1-1: Comparison of experimental readings and modelling results

No	Data Input				Output											
	Hot Water Inlet		Cold Water Inlet		Hot Water Out Temp.				Cold Water Out Temp.				LMTD			
	Temp.	$\dot{V}$	Temp	$\dot{V}$	Model	Reading	Error		Model	Reading	Error		Model	Reading	Error	
	°C	l /min	°C	l / min	°C		K	%	°C		K	%	K		K	%
1	20.5	7.2	15.4	1.0	20.3	19.9	0.4	2.0	16.6	16.3	0.3	1.8	4.36	4.31	0.05	1.3
2	21.1	7.2	15.3	4.0	20.8	20.6	0.2	1.0	16.4	15.9	0.6	3.8	5.07	5.23	0.16	3.1
3	29.9	7.2	15.3	1.0	29.3	29.2	0.1	0.3	20.1	19.8	0.3	1.5	11.7	11.8	0.1	1.0
4	29.8	7.2	15.0	4.0	28.9	28.7	0.2	0.7	17.6	17.0	0.6	3.5	13.0	13.2	0.2	1.7
5	40.0	7.2	16.0	1.0	38.8	37.7	1.1	2.9	24.6	24.8	0.2	0.8	18.8	18.2	0.6	3.2
6	39.9	7.2	15.4	4.0	38.7	38.2	0.5	1.3	18.3	19.0	0.7	3.7	22.4	21.7	0.7	3.0
7	56.3	7.2	17.0	1.0	54.1	55.0	0.9	1.6	32.0	33.2	1.2	3.6	29.9	29.7	0.2	0.6
8	56.1	7.2	16.3	4.0	53.0	53.6	0.6	1.1	21.7	22.6	0.9	4.0	35.4	35.2	0.2	0.5

The average temperature deviations and LMTD errors of above eight comparisons were 0.37K and 1.6%, correspondingly. The higher errors occurred in No.2, 5 and 6 comparisons. The close temperature difference of hot water and cold water inlet temperatures and temperature fluctuation during the experiments might have resulted in the inaccuracy of both measurement and modelling. However, the overall results from single panel modelling overall were quite reasonable.

Based on these comparisons, the temperature differential condition (already referred to) was added to the heat exchanger programme for improvement. When sensors read a temperature difference between hot water inlet and cold water inlet less than 5K, the programme would consider the heat transfer to be insignificant and the cold water circulation pump immediately stopped until the temperature difference was above 5K again.

6.2 Discussion of Simulation Results

6.2.1 Analysis on Multi-Panel Heat Exchanger

In this section, the effect of the number of panels on heat exchanger performance was analysed through the results of modelling. A computer programme was run with an initial cold circulating water flow rate of 4 litre/min and the gathered experimental data of the warm waste water from the dishwasher. In the programme, the cold water inlet temperature was the outlet temperature of mixed water in the mixing chamber of the hot water storage heater. The simulation result of the multi-panel heat exchanger is shown in Figure 6.2-1. The red line (Panel 1 In) is the input of the original experimental temperature data to the model. The temperature increase of cold water on each panel is presented in Figure 6.2-2.

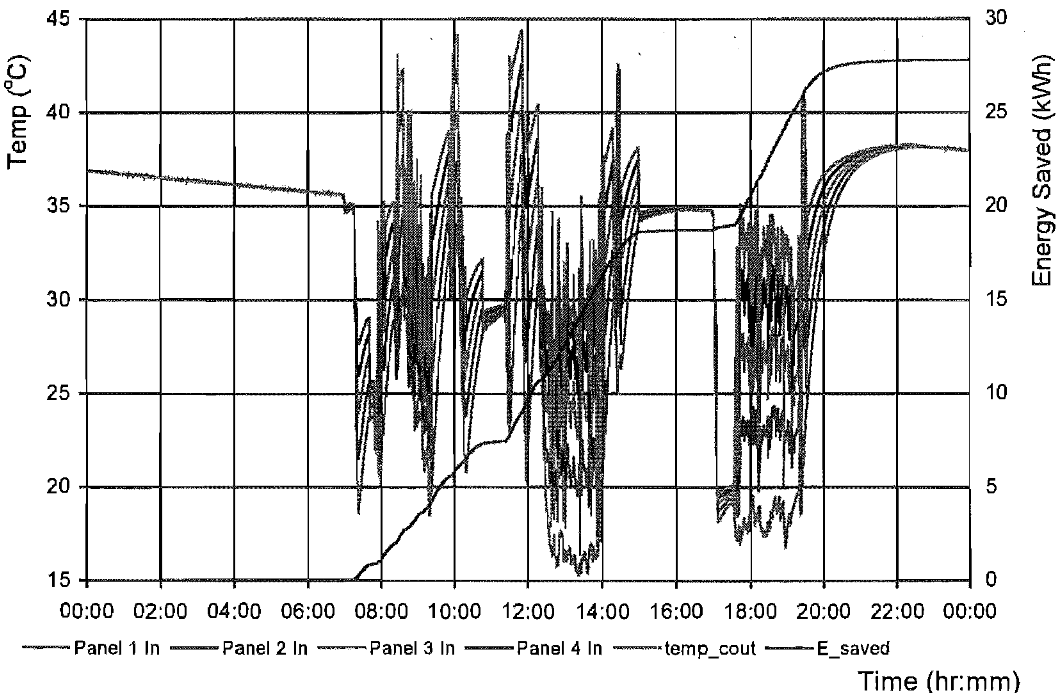


Figure 6.2-1: Temperature increase on a four-panel heat exchanger modelling

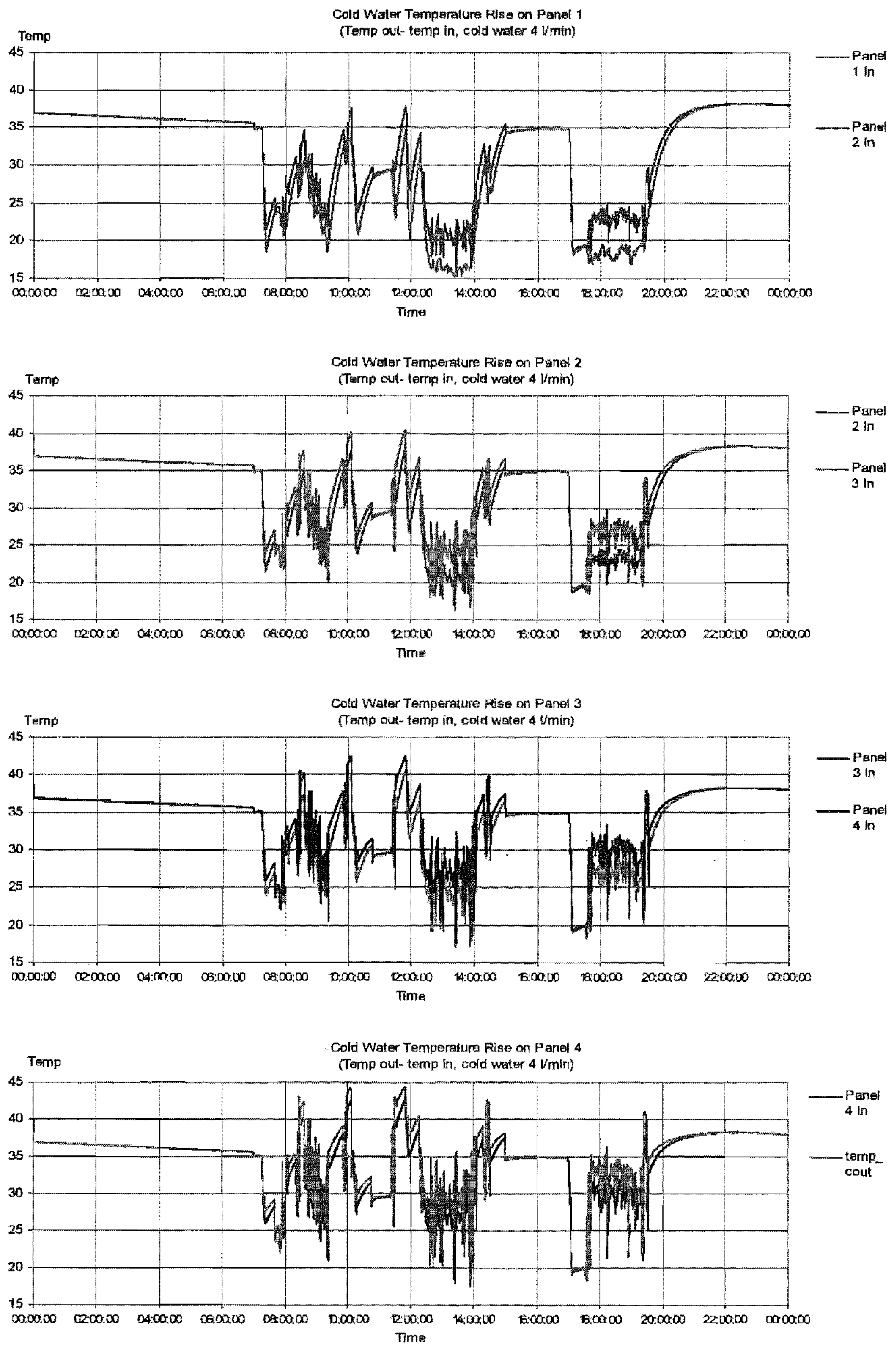


Figure 6.2-2: Temperature increase of cold water on each panel

6.2.1.1 Temperature Increase of Cold Water on Each Panel

Table 6.2-1: Temperature Increase of Cold Water during a period of Lunch Time

No.	Time	Hot In Temp °C	Temp In, °C Panel No.				Panel 4 Out C	Temp Rise of Cold Water K Panel No.				Overall Increase K
			1	2	3	4		1	2	3	4	
1	12:39:09	49.4	16.3	23.0	28.0	31.7	34.7	6.7	5.0	3.8	3.0	18.4
2	12:40:39	41.5	17.0	20.0	22.6	24.9	26.9	3.0	2.6	2.3	2.0	10.0
3	12:42:09	48.5	17.4	19.8	22.0	24.0	25.9	2.4	2.2	2.0	1.9	8.5
4	12:43:39	36.4	17.2	20.3	22.9	25.1	26.9	3.1	2.6	2.2	1.8	9.7
5	12:45:09	37.9	16.2	20.0	23.0	25.5	27.5	3.8	3.0	2.5	2.1	11.3
6	12:46:39	39.8	15.9	20.2	23.5	26.2	28.5	4.3	3.3	2.7	2.3	12.6
7	12:48:09	47.9	15.8	21.8	26.4	30.0	32.9	6.0	4.6	3.6	2.9	17.1
8	12:49:39	49.2	16.7	22.5	27.0	30.6	33.5	5.7	4.5	3.6	2.9	16.8
9	12:51:09	48.2	17.5	23.5	28.0	31.5	34.3	6.0	4.5	3.5	2.8	16.8
10	12:52:39	40.4	17.5	21.6	24.7	27.3	29.5	4.1	3.2	2.6	2.2	12.0

Table 6.1-1 lists the programming results of the cold water temperatures and the temperature increases between the inlet and outlet of each panel which were abstracted from figures shown above during a period of lunch time.

The average temperature increases of the above 10 samples are 4.2K, 3.4K, 2.9K, 2.5K for panels 1 , 2, 3 & 4. The temperature increase on each panel declines with the increase of the panel number, as anticipated in the discussion of Section 5.2.

The largest temperature increase of cold water on the first panel also means that it transfers the largest amount of heat and thus has the highest heat transfer efficiency while the last panel has lowest efficiency among all panels. As the initial cost of a thermosyphon heat exchanger is related to the number of thermosyphon panels that are installed in the heat exchanger, the more panels that are installed, the greater the cost of a heat exchanger. Therefore those thermosyphon panels which achieve a lower temperature increase on the cold water side are not so effective from the economic point of view. In the simulation result, the fourth panel has an average temperature rise of 2.5 K. The fifth panel would have only about 2K increase if it were installed, which is too small and possibly not economical as this 2K might be achieved by using other ways which will be discussed in Chapter 7.

6.2.1.2 Overall Temperature Increase

Of all 2880 samples programmed every 30 seconds for a 24-hour period, 21.8K is the maximum temperature increase of cold water for the proposed 4-panel thermosyphon heat exchanger. This occurred at 19:23:39, when the hot water in the pre-wash tank of the dishwasher was dumped into drainage.

From the figures in this chapter, it can be seen that sharp temperature increase curves happened at the beginning and the end of each dishwashing. At the beginning, hot waste water started to flow into the heat exchanger which caused a sudden switch-on to the panels. At the end of a dishwashing process, a large amount of hot water was drained to the heat exchanger which made all panels work on full load.

6.2.1.3 Economic Feasibility

In a sample day, the energy saved on a 4-panel thermosyphon heat exchanger would be 27.8 kWh. If excluding 3 months school holiday, the overall electricity that could be saved would be 27.8 kWh/day×30 day/month×9month=7500 kWh.
Based on an electricity rate of $0.206 per kWh (Christchurch residential rate, Meridian Energy), the predicted annual cost saving would be $1500.

6.2.2 Analysis on Hot Water Storage Tank and Heat Recovery System

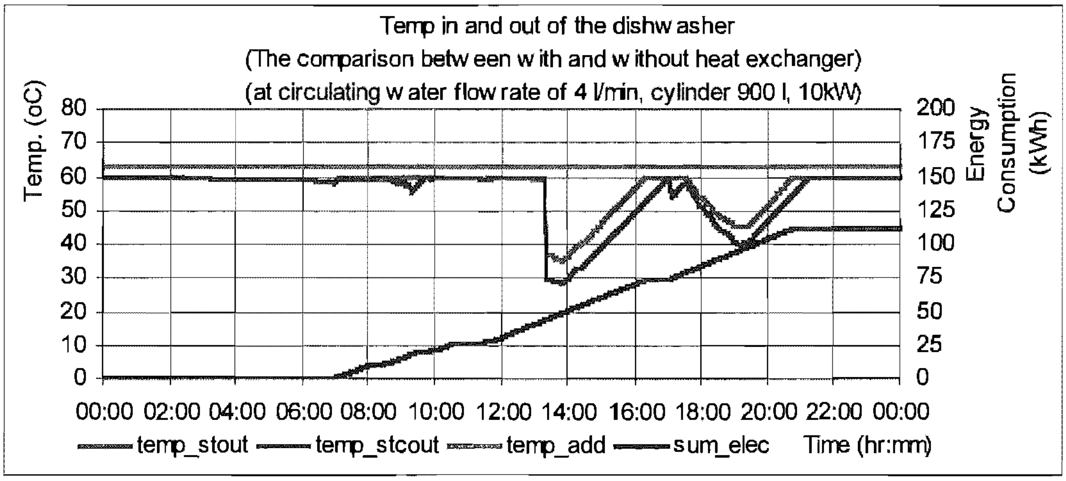
6.2.2.1 Temperature Output vs. Heating Element Power Input

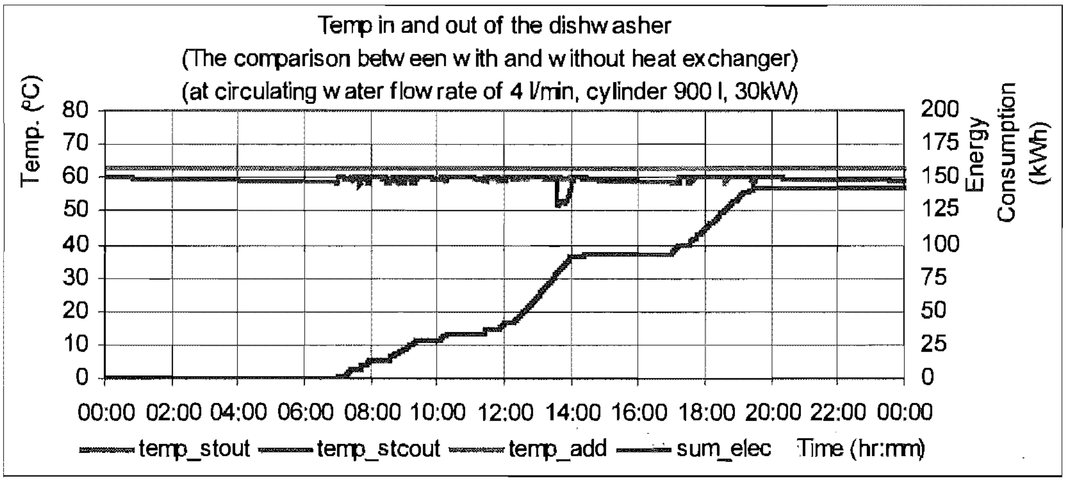
Figure 6.2-3 shows three hot water profiles when the power inputs to hot water storage heater were 10 kW, 30 kW, 50 kW from the top to the bottom. The red line in the figures represents the outlet temperature from the hot water storage heater when a 4-panel thermosyphon heat exchanger is employed, whereas the blue line is the outlet temperature without the heat exchanger. The pink line is the temperature of the dishwasher pre-wash tank where a supplementary heat element is installed to heat the pre-wash water to a temperature of 63°C, according to previous data collection from the dishwasher at the University Halls of Residence. The black line in the figure is the energy consumption profile of the dishwasher in a day.

For a 900litre and10 kW power input hot water storage heater, the blue line remained constant until 8am when morning dishwashing started. This temperature dropped from 60°C to 55°C and rose again to 60°C again at 9:30 am when the dishwashing finished. Meanwhile, the red line remained unchanged. This implies that while the heating capacity in the cylinder is not sufficient for hot water supply, the energy recovered by heat exchanger has compensated for this energy shortage so that the temperature out of the hot water tank could still maintain the required temperature (60°C). At lunch, a sudden drop occurred at 12:40. The hot water outlet temperature fell to 30°C in a short period and did not fully return to 60°C until 21:00. Employing the thermosyphon heat exchanger shortened the time for temperature recovery. However, the power input of the hot water storage heater was still too small to recover temperature in time so more supplementary heating was needed to fill up the gap between the low temperature out of hot water cylinder and the pre-wash temperature set-up of the dishwasher (63°C).

With the capacity of heating element in the hot water tank increased to 30kW, when the proposed heat exchanger was installed, the water draw-off temperature of the hot water tank remained at the setting temperature 60°C, whereas a temperature drop occurred for half an hour when the heat exchanger was not installed.

With the hot water tank power input increased further to 50kW, water draw-off remained constant at the temperature of 60°C without the help of the dishwasher supplementary heating, which means the water tank heating capacity was sufficient to satisfy the requirements.





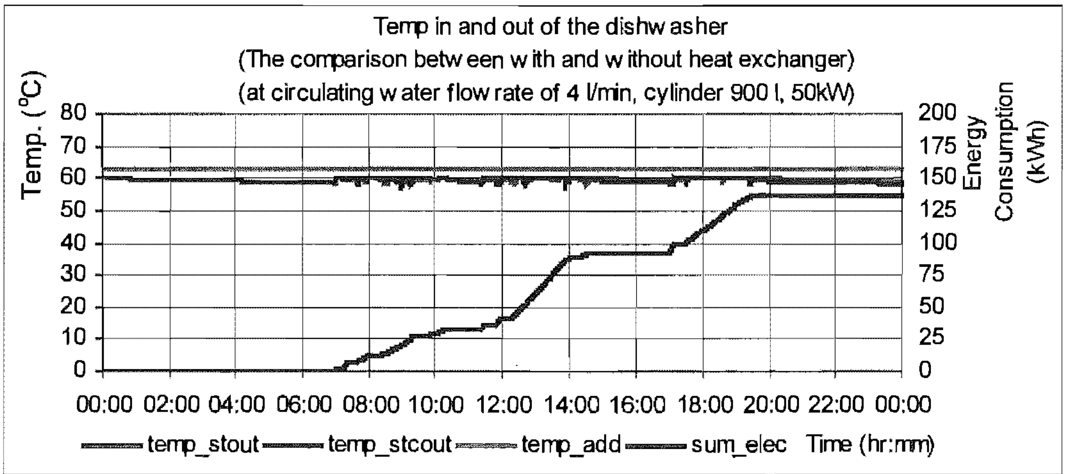


Figure 6.2-3: Hot water temperature profiles of dishwasher at different power inputs

6.2.2.2 Hot Water Tank Power Input in and Dishwasher Supplementary Heating

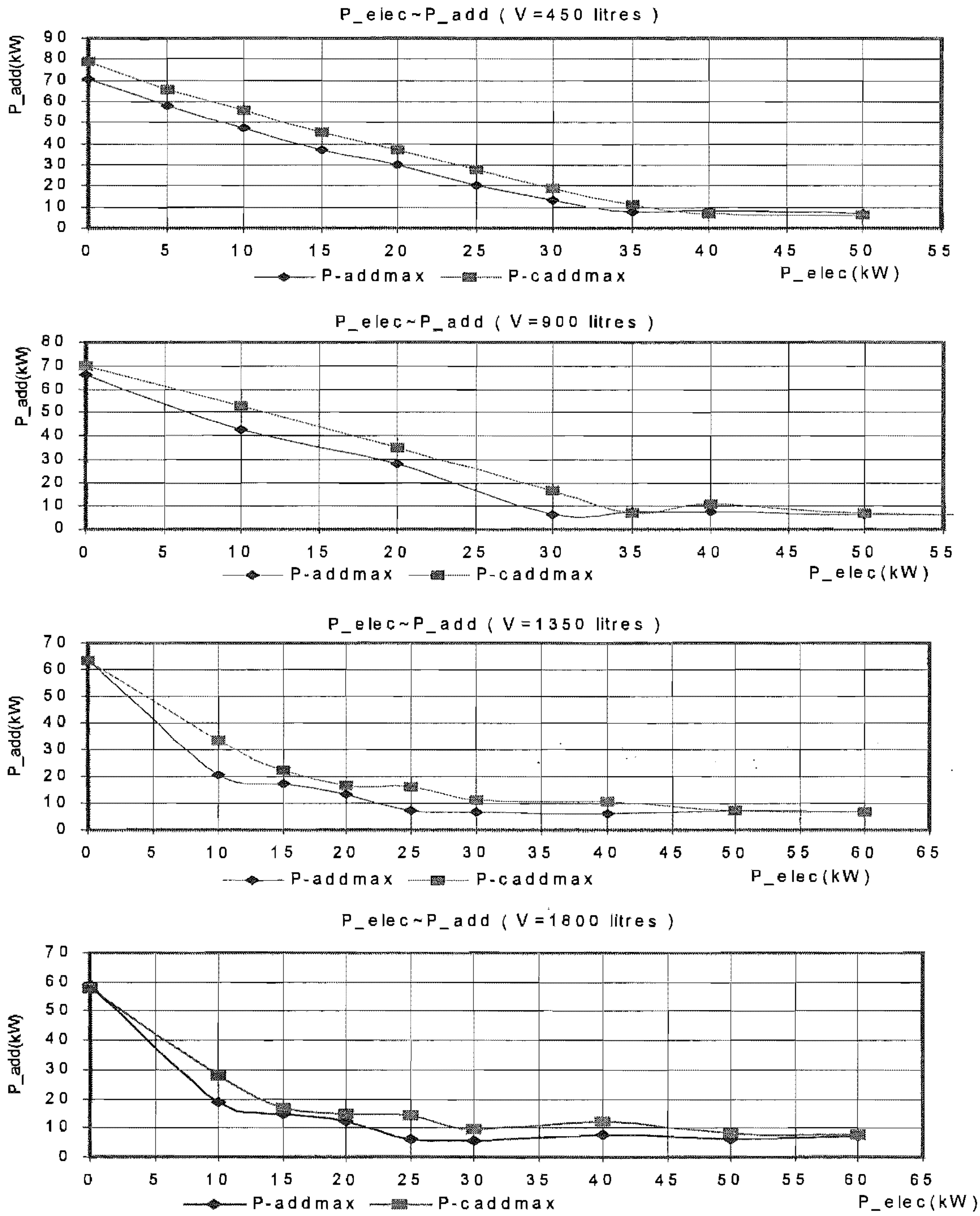


Figure 6.2-4: A comparison of hot water tank heating and supplementary heating capacities with and without heat exchanger

Figure 6.2-4 is a comparison of the hot water tank heating and supplementary heating capacities needed when there is the proposed heat exchanger and there is no heat exchanger in the dishwasher system at various sizes of hot water cylinder from 450 litre to 1800 litre.

In the figure, the x-axis is P_elec which represents the hot water tank power input, while the y-axis is the supplementary power input. The blue line P_addmax is the minimum supplementary power input needed to achieve the temperature settings with the proposed heat exchanger, whereas, the Red line P_caddmax is the minimum supplementary power input with no heat exchanger installed.

In all four charts, when there is no power input on the hot water cylinder, the dishwasher system needs the highest minimum supplementary heating capacity in an effort to reach design dishwashing temperature. However, when the system has the heat exchanger the supplementary power needed is smaller than that when system has no heat exchanger at the same hot water tank power input.

Supplementary power input decreases with the increase of hot water tank power until the tank heating input reaches a point at which the supplementary power input lines flatten out. This point could be considered as a minimum overall heating capacity needed for the system. Beyond that point supplementary heating remains constant even when the heating capacity increases. It could be interpreted that the existing power input on hot water cylinder is large enough to reach 60°C under the water draw-off pattern recorded. The constant supplementary power is for heating water from 60°C to the dishwasher required 63°C.

6.2.2.3 Optimal Hot Water Tank Power Input

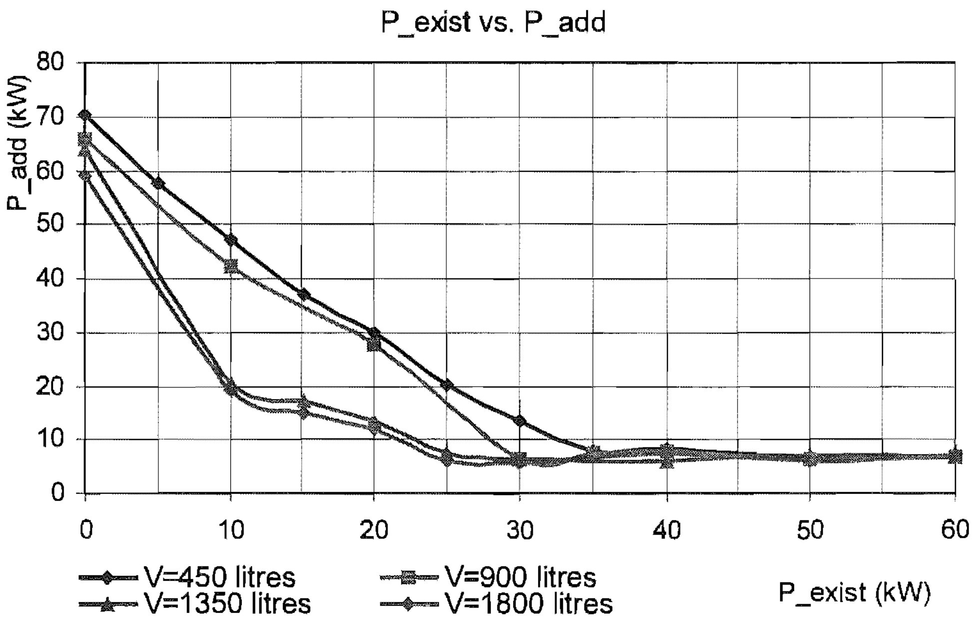


Figure 6.2-5: Existing water tank power input vs. supplementary power input

(with heat exchanger)

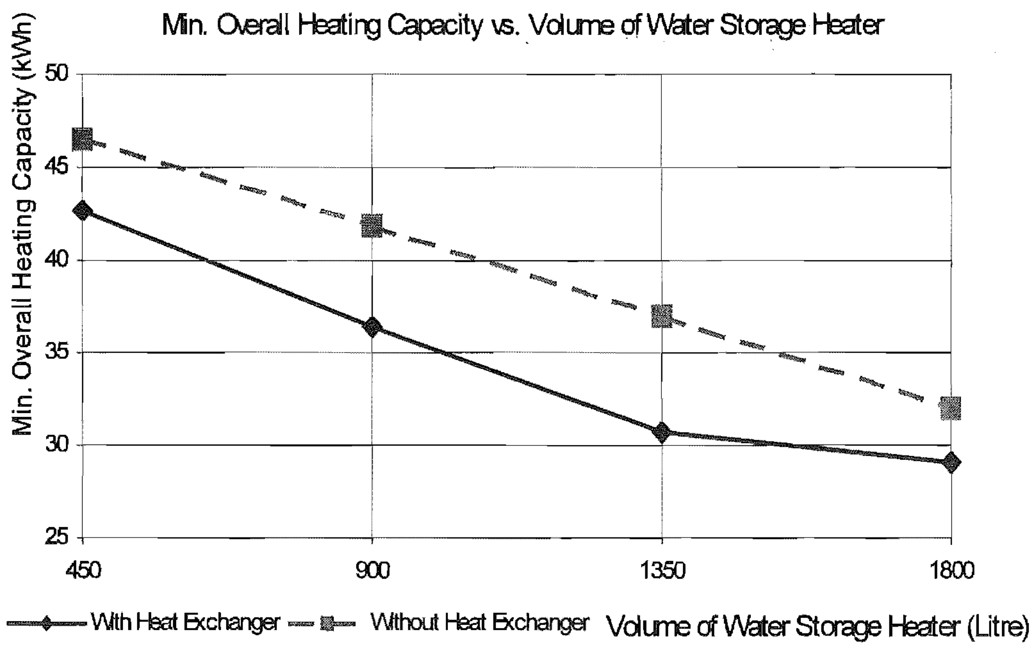


Figure 6.2-6: Minimum overall heating capacity vs. hot water cylinder size

Figure 6.2-5 shows the influence of cylinder size on the hot water tank power and supplementary power inputs in this particular heat exchanger recovery system. In the figure it can seen that the top two lines (450litre and 900litre lines) are close together while the bottom two lines (1350litre and 1800litre) behave essentially the same. However from some points all four lines are drawn together. We also find in Figure 6.2-5 and Figure 6.2-6 that when the cylinder size is between 1350 and 1800 litres, the minimum overall heating capacity changes slightly from 31kW to 29kW. It implies that if the cylinder size is over 1350 litres, the minimum overall power needed in the system is about 31kW, comparing to 43kW and 37kW when the cylinder size is 450 litres and 900 litres, correspondingly. As a result, 1350litre and 31kW can be regarded as the optimum volume capacity for the cylinder and heating capacity for the whole system if the system has the proposed thermosyphon heat exchanger.

Figure 6.2-4 and Figure 6.2-5 also show a phenomenon in which four heating capacity curves break into two sets, with almost identical characteristics for 450 and 900 litres as one set, and 1350 and 1800 litres as the other set. In the former set, each curve consists of two reasonably straight lines, one sloped and one flat, with an obvious turning point link between. In the latter set, each curve is more like formed by three lines, two sloped and one flat although the curve slightly fluctuates before turning flat and it has no obvious turning point. On the middle sloped lines of 1350 and 1800 litre curves, the overall power inputs of the system (sum of the x-axis and y-axis values) are nearly constant at 31kW as was discussed in last paragraph. It could be explained that the optimal hot water power input is a critical point (35kW and 30kW for the 450 and 900 litre curves correspondingly), and a critical line for the 1350 and 1800 litre set. In another words, when the hot water tank power input is between 10kW and 30kW, the overall power consumption of the system remains optimal.

Chapter 7: CONCLUSIONS AND RECOMMENDATIONS

7.1 Introduction

The objective of this research was to develop a multiple panel thermosyphon heat exchanger for a waste water heat recovery system. There were three aspects in the research. Firstly, experiments were carried out to determine the performance characteristics of a single thermosyphon panel. Secondly, measurements of hot water usage and waste water temperature and flow rates were obtained for a potential application of the proposed exchanger (the dishwasher for the kitchen in the University Halls of Residence). Finally, a model of a multi-panel thermosyphon heat exchanger was developed to predict the energy savings that would be expected if such a heat exchanger was used in this situation.

7.2 Thermal Performance Of A Single Thermosyphon Panel

A comparison between the overall heat transfer coefficients with top surface insulated and uninsulated in the same working conditions showed that the bottom surface of the panel contributed most of heat transfer while little heat were transferred through the top surface of the panel. It could be explained that for a multi-panel thermosyphon heat exchanger, even when the top surfaces of panels are covered by deposits from the waste water it would likely to have little effect on thermal performance of the heat exchanger

For a thermosyphon panel, theoretically the overall heat transfer coefficient U should be dependant on the total surface area of the top and bottom sides of the panel. However, Figure 3.4-6 shows that under the same working condition (cold water inlet temperature was 25°C and its volumetric flow rate varied from 4, 6, 8 to 10 litre/min), the overall heat transfer coefficients were 985, 1045, 1010, 1025 $W/m^2.K$ respectively for the top surface insulated panel, and 985, 1005, 1005, 1060 $W/m^2.K$ correspondingly for the same but the top surface uninsulated panel. By the comparison, it could be seen that the overall heat transfer coefficient U was dominated by the bottom side of the panel. As the U values were based on the bottom surface only, it was desirable to operate the panel with a slope that was as small as possible to maximise the contact area of working fluid and the sheet plate of the panel. The inclination angle test has proved that when the single panel heat exchanger was horizontal, it was still able to transfer heat from hot water to cold supply although the amount was limited. The inclination angle of 10° could be identified as the minimum inclination angle at which good performance was still obtained.

The advantages of the proposed heat recovery system are:

1) Simplicity of construction;

2) The proposed multi-panel thermosyphon heat exchanger should be low maintenance because of insensitivity to fouling;

3) The heat exchanger is able to recover energy and store the recovered heat in the hot water storage tank even when hot water not being used;

4) The drain pipe installed on the bottom of the heat exchanger makes it easy to dump waste water from the heat exchanger when useful energy recovery is completed.

7.3 Data Acquisition From A Dishwasher System

Key aspects of the water usage data of a dishwasher system collected from the cafeteria in the University Halls of Residence during Lunch Time are:

1) The average daily hot waste water discharge for the dishwasher was 2500 litre/day;

2) The pre-wash temperature of the dishwasher was between 48°C and 63°C;

3) The temperature range of the discharged waste water was between 35°C and 55°C.

The dishwasher data collected indicated that the dishwasher system had a desirably high waste water discharge temperature. It fell within the working temperature range of the single panel heat exchanger in the single panel component test. The data from the simple dishwasher system would enable heat transfer calculations to be performed for the model simulation of the proposed thermosyphon heat exchanger system being used in conjunction with this dishwasher.

7.4 Simulation Of A Multi-Panel Thermosyphon Heat Exchanger

The results based on a day's data input for a 4-panel thermosyphon heat exchanger used in conjunction with the dishwasher system in the Halls of Residence are:

1) The average temperature increase on each panel (from panel 1 to panel 4 according to the direction of the cold water flow) is: 4.2 K, 3.4 K, 2.9K, 2.5K, correspondingly.

2) When the volume capacity of the hot water storage tank was 900 litres, the maximum overall temperature increase of the cold water on the thermosyphon heat exchanger is 21.8 K.

3) The predicted energy saved would be 27.8 kWh/day for the thermosyphon heat exchanger heat recovery system.

4) The optimum combination of the capacity of the hot water cylinder element plus the capacity of the booster heater in the dishwasher when the system includes the HX was 31 kW when the volume capacity of the hot water storage heater was 1350 litres.

7.5 Recommendations for future research

It is recommended that future research of the thermosyphon heat exchanger should focus on the following areas:

1) Increase cold water tube heat transfer surface area

   Comparing with the large surface area of the panel ($0.65m^2$), the heat transfer surface area of the cold water tube is very small ($0.05m^2$). This might cause "Bottle Neck" for the cold water tube to take away as much heat as the working fluid is capable to absorb. Some options are available for increasing the heat transfer surface area of the cold water tube by changing the geometry of the cold water tube, such as thin fins, coiled tubes..., and hence enhancing the internal and external heat transfer.

2) Size of the multi-panel heat exchanger

   A further study on the sizing of the heat exchanger could be carried out for a better performance of the heat exchanger, such as the number of the panels and the volume of the heat exchanger tank. An appropriate channel height between two parallel panels could sufficiently utilize the space of the tank and recover the heat from the waste water to the utmost. Besides, a compact tank would be more economical when hot waste water discharge is not large. However, the number of the panels and the heat exchanger size are greatly dependant on the hot water usage.

3) Costing of heat exchanger

Finally, and most importantly, an investigation on the costing of producing such a multi-panel thermosyphon heat exchanger should be undertaken to evaluate its economic feasibility.

Depending on the outcome of the above investigations into possible design enhancements, and particularly the costing, a prototype heat exchanger could be built and its short and long term performance evaluated in the actual working environment in which it would be required to operate. This would be the ultimate test of whether or not the concept has a future as a means of achieving a worthwhile and economic saving of energy.

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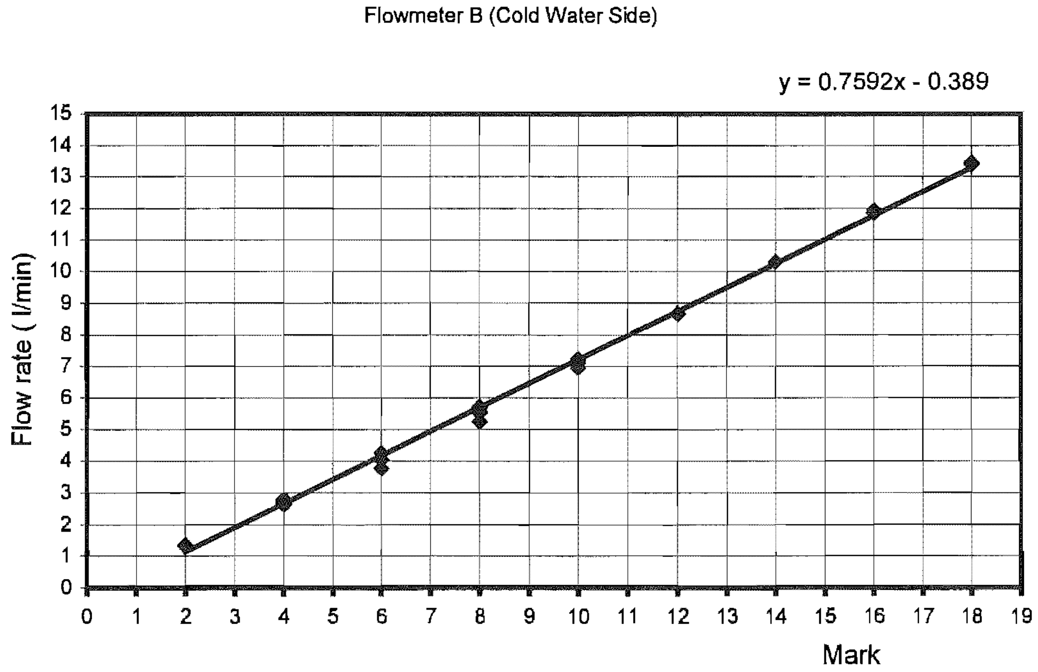
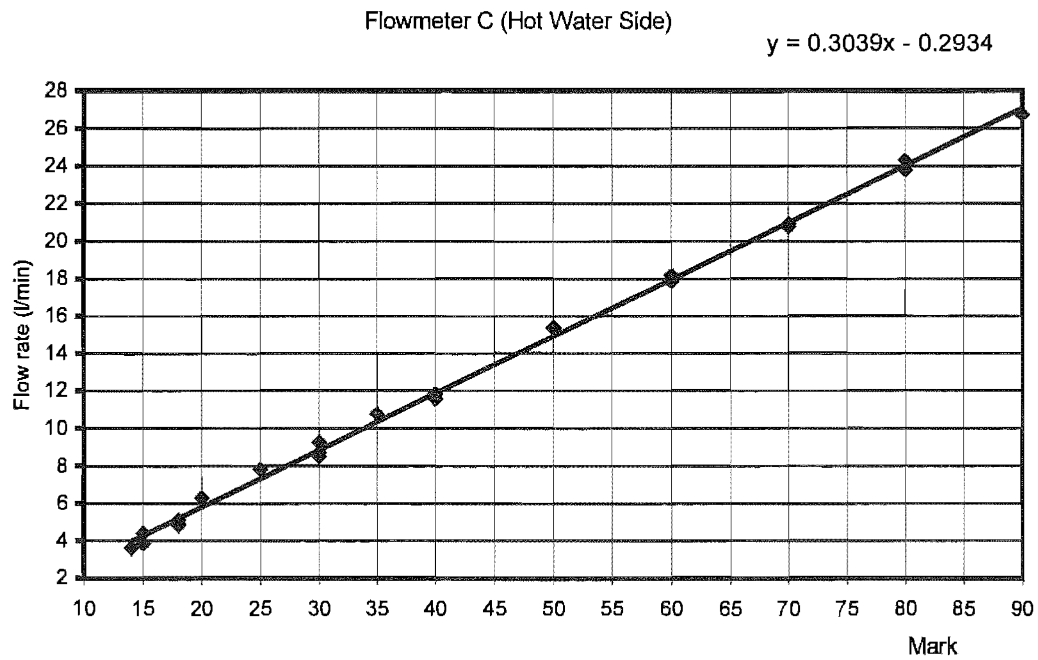
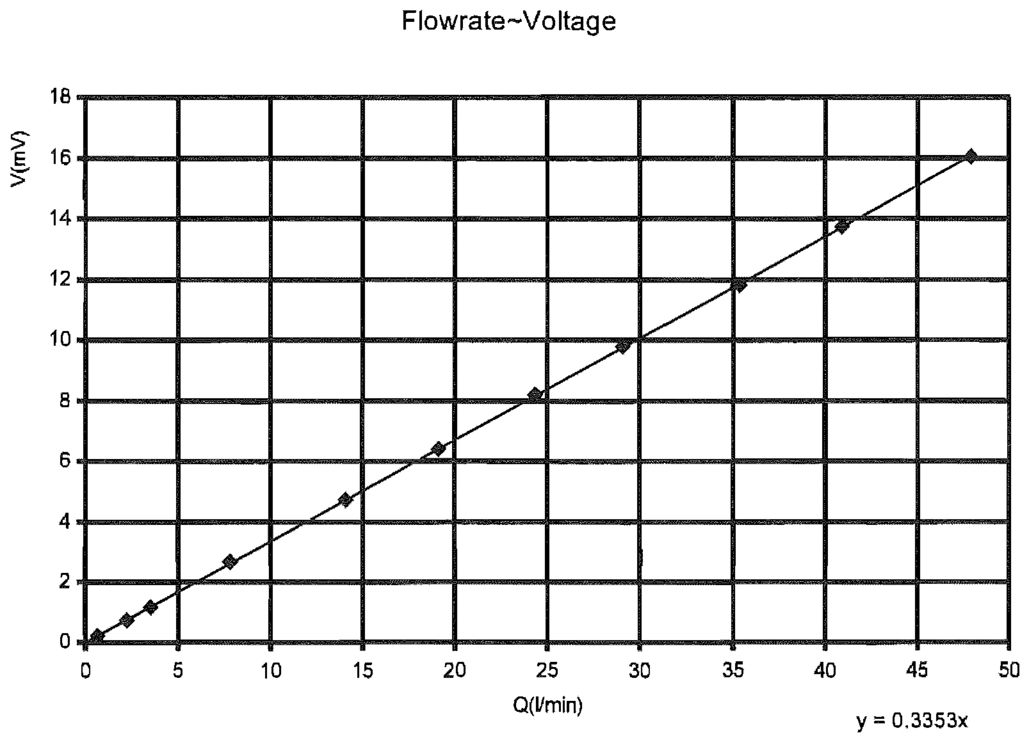
Appendix A: Flowmeter Calibration

A1. Flowmeters B and C





A2. CPOA-XE/MAG-XE Flow Meter Calibration



APPENDIX B: PROGRAMME LISTING

B.1 SIMULATION ON MULTI-PANEL HEAT EXCHANGER AND ITS HEAT RECOVERY SYSTEM

```
%Purpose:
% To simulate heat transfer process on a proposed heat exchanger and its heat recovery system;

%Initialize simbols:
number=4
L=0.97
W=0.9
distance=0.05
H=0.26+(number-1)*distance
area=L*W
V_exch=L*W*H
V_panel=0.915*0.736*0.005
V_water=V_exch-V_panel*number
pump=100
k_poly=0.04
A=2*(L*H+L*W+H*W)
thk=0.035

fprintf('This program is to simulate the heat transfer process on a thermosyphon heat exchanger.\n','s')

inputfile=input('Input data file collected from the dishwasher=:','s')
t=input('Input time interval of the computer data in seconds=:\n')
flow_in=input('Input flow rate of the cold circulating water, litre/minute=:\n')

n=input('Number of rows of data input samples=:\n')
V_heater=input('Input the volume capacity of the hot water heater, in litres=:\n')
P_elec=input('Input the capacity of the heating element, in kW=:\n')
percent=input('Input the percentage of the volume of the mixing chamber=:\n')

%Heat loss of the hot water heater (NZS4602: 1988)
if V_heater<90
  heat_loss=0.0084*V_heater+0.4
else
  heat_loss=0.0048*V_heater+0.72
end

Elec=P_elec*t
V_mix=percent*V_heater
D_heater=0.6
```

```
H_thermo=0.2
V_ther=1000*pi*(D_heater^2)*H_thermo/4
V_stra=(1-percent)*V_heater
loss=heat_loss*percent*t/24
loss_ther=heat_loss*V_ther*t/(V_heater*24)
loss_stra=heat_loss*t/24-loss-loss_ther
temp_diff=1

%Load input file:
[data, headertext]=xlsread(inputfile);
temp3=data(1,5)
temp6=data(1,8)
flow8=data(1,11)
flow9=data(1,13)
temp_wstd=temp3

for i=1:5
        %initial values, for ii=1 only
        temp_cstd=temp_wstd-temp_diff
        temp_hstd=60
        temp_ststd=60
        temp_hcstd=60
        temp_stcstd=60
        sum_E=0;
        sum_pump=0
        sum_elec=0
        ssum_elec=0
        V=V_ther
        VV=V_ther
        Elec_pre=0
        EElec_pre=0

        %Loop to read input files:
        for ii=1:n
      date=data(ii,1)
      time=data(ii,2)
      temp11=data(ii,3)
      temp22=data(ii,4)
      temp33=data(ii,5)
      temp44=data(ii,6)
      temp55=data(ii,7)
      temp66=data(ii,8)
      temp77=data(ii,9)
      flow88=data(ii,11)
      flow99=data(ii,13)
      flow_cold=flow99
```

```
temp_cold=temp66
temp_cent=temp_cstd

clear loss_exchanger
loss_exchanger=0.001*k_poly*A*(temp_wstd-temp55)*t/thk
heatrate=k_poly*A*(temp_wstd-temp55)/thk
out1(ii,1)=data(ii,1)
out2(ii,1)=data(ii,2)

% PART 1: HEAT EXCHANGER CALCULATION
if flow88<=0
   flow_went=0
   temp_went=temp_wstd
   %Loop for panel performance calculation
   sum_tempwout=0
   sum_stagesave=0
   for jj=1:number
      delta_temp=temp_wstd-temp_cent
      delta_temp=min(40,delta_temp)
      if delta_temp<temp_diff
         fprintf('This panel is not working.')
         flow_cent=0
         QQ=0
         EE=0
         EE_pump=0
         temp_cout=temp_cent
         T=temp_wstd
         [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
         temp_wout=temp_wstd-loss_exchanger/(V_water*rho*cp)
      else
         clear T cp rho mu nu k alpha beta Pr
         T=temp_cent
         [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
         flow_cent=flow_in
         flow_cin=flow_cent/60
         c_cold=flow_cin*rho*cp
         c_hot=0
         cmin=c_cold

         %Import function to calculate overall heat transfer coefficient
         u=overcoeffi(flow_cent,delta_temp)
         ntu=u*area/cmin
         ntu=min(10,ntu)
         effect=-2.39E-4*ntu^3+8.44E-4*ntu^2+6.68E-2*ntu
         QQ=effect*cmin*(temp_wstd-temp_cent)/1000;   % in kW
         EE=QQ*t/(60*60)
```

```
        EE_pump
        if  QQ<=0
           fprintf('QQ is negaitive, temp_went is lower than temp_cent\n','s')
           break
        else
        end
        clear T cp rho mu nu k alpha beta Pr
        T=temp_wstd;
        [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
        temp_wout=temp_wstd-(QQ*t)/(V_water*rho*cp/(1+number))-
        loss_exchanger/(V_water*rho*cp)
        temp_cout=QQ*1000/c_cold+temp_cent
        outb(ii,jj)=cmin
        outc(ii,jj)=u
        outd(ii,jj)=ntu
        oute(ii,jj)=effect
     end
     sum_E=sum_E+EE
     temp_wout=max(temp_wout,temp55)
     sum_tempwout=sum_tempwout+temp_wout
     sum_stagesave=sum_stagesave+EE
     pump(ii,jj)=EE_pump
     flow(ii,jj)=flow_cent
     outa(ii,jj)=delta_temp
     out3(ii,jj)=QQ
     out4(ii,jj)=sum_E
     out9(ii,jj)=temp_wout
     out12(ii,jj)=temp_cent
     out13(ii,jj)=temp_cout
     temp_cent=temp_cout; %reset for j+1 loop
  end

  clear EE_pump flow_cent
  EE_pump=max(pump(ii,1:number))
  flow_cent=max(flow(ii,1:number))
  clear T cp rho mu nu k alpha beta Pr
  T=temp_wstd
  [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
  temp_wtop=temp_wstd-loss_exchanger/(V_water*rho*cp)
  temp_wtop=max(temp_wtop,temp55)
  sum_pump=sum_pump+EE_pump
  temp_wmix=(sum_tempwout+temp_wtop)/(1+number)

  %Create results output
  out5(ii,1)=sum_pump
  out6(ii,1)=flow_went
```

```
        out7(ii,1)=temp_went
        out8(ii,1)=temp_wstd
        out10(ii,1)=temp_wmix
        out11(ii,1)=flow_cent

    else
        temp_went=temp33
        flow_went=flow88/(number+1)
        sum_tempwout=0
        sum_stagesave=0

        [cp_wstd,rho_wstd,temp_went,cp_went,rho_went,tempin_wstd,
        cpin_wstd,rhoin_wstd]=mixed(t,V_water,temp_wstd,temp_went,flow_went)
        for jj=1:number
                delta_temp=tempin_wstd-temp_cent
                delta_temp=min(40,delta_temp)
            if delta_temp<temp_diff
                fprintf('This panel is off.')
                flow_cent=0
                QQ=0
                EE_pump=0
                EE=0
                temp_cout=temp_cent
                T=temp_wstd
                [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
                temp_wout=temp_wstd-loss_exchanger/(V_water*rho*cp)
            else
                clear T cp rho mu nu k alpha beta Pr
                T=temp_cent
                [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
                cp_cent=cp
                rho_cent=rho
                flow_cent=flow_in
                flow_cin=flow_cent/60
                c_cold=flow_cin*rho_cent*cp_cent
                c_hot=flow_went*rhoin_wstd*cpin_wstd
                cmin=min(c_hot,c_cold)
                %Import function to calculate overall heat transfer coefficient
                u=overcoeffi(flow_cent,delta_temp)
                area=L*W
                ntu=u*area/cmin
                ntu=min(10,ntu)
                effect=-2.39E-4*ntu^3+8.44E-4*ntu^2+6.68E-2*ntu
                QQ=effect* cmin*(tempin_wstd-temp_cent)/1000
                EE=QQ*t/(60*60)
                EE_pump=100*0.001*t/3600
```

```
            if  QQ<=0
               fprintf('sum_QQ is negaitive, temp_went is lower ...
                       than temp_cent\n','s');
               break
            else
            end
            clear T cp rho mu nu k alpha beta Pr
            T=tempin_wstd
            [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
            temp_wout=tempin_wstd-(QQ*t)/(V_water*rho*cp/(1+number))-
            loss_exchanger/(V_water*rho*cp)
            temp_cout=QQ*1000/c_cold+temp_cent
            outb(ii,jj)=cmin
            outc(ii,jj)=u
            outd(ii,jj)=ntu
            oute(ii,jj)=effect
         end
         sum_E=sum_E+EE
         temp_wout=max(temp_wout,temp55)
         sum_tempwout=sum_tempwout+temp_wout
         sum_stagesave=sum_stagesave+EE
         pump(ii,jj)=EE_pump
         flow(ii,jj)=flow_cent
         outa(ii,jj)=delta_temp
         out3(ii,jj)=QQ;
         out4(ii,jj)=sum_E
         out9(ii,jj)=temp_wout
         out12(ii,jj)=temp_cent
         out13(ii,jj)=temp_cout
         temp_cent=temp_cout
      end

      clear EE_pump flow_cent
      EE_pump=max(pump(ii,1:number))
      flow_cent=max(flow(ii,1:number))
      clear T cp rho mu nu k alpha beta Pr
      T=temp_wstd
      [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
      temp_wtop=tempin_wstd-loss_exchanger/(V_water*rho*cp)
      temp_wtop=max(temp_wtop,temp55)
      sum_pump=sum_pump+EE_pump
      temp_wmix=(sum_tempwout+temp_wtop)/(1+number)
      %Create result output
      out5(ii,1)=sum_pump
      out6(ii,1)=flow_went
      out7(ii,1)=temp_went
```

```
        out8(ii,1)=tempin_wstd
        out10(ii,1)=temp_wmix
        out11(ii,1)=flow_cent
    end

    temp_wstd=temp_wmix
    clear temp_went temp_wout temp_cent temp_cout
    temp_cent=temp_cstd
    temp_cout=out13(ii,number)
    [temp_cmix]=cmix(t,temp_cent,temp_cout,temp_cold,temp_cstd,flow_cent,
    flow_cold,V_mix,loss)
    out14(ii,1)=flow_cold
    out15(ii,1)=temp_cold
    out16(ii,1)=temp_cstd
    out17(ii,1)=temp_cmix
    result1=[out1,out2]
    result2=[out3,out4,out5]
    result3=[out6,out7,out8,out9,out10]
    result4=[out11,out12,out13]
    result5=[out14,out15,out16,out17]
    temp_cstd=temp_cmix

  diff_wstd=abs(out8(2880,1)-out8(1,1))
  diff_cstd=abs(out16(2880,1)-out16(1,1))
  if diff_wstd<=1 & diff_cstd<=1
    fprintf('i=%d\n',i);        end
    fprintf('The End of the Programme.\n');
    break
  else

    clear out1:17
  end
end

            temp_wstd=(out8(2880,1)+out8(1,1))/2;
```

B.2. SIMULATION ON HOT WATER STORAGE HEATER AND DISHWASHER

```
%Purpose:Simulation of hot water storage tank and a dishwasher system.
inputfile=input('Input data file obtained from heat exchanger calculation=:','s')
V_heater=input('Input the volume capacity of the hot water heater, in litres=:\n')
P_elec=input('Input the heating element capacity of the hot water heater, in kW=:\n')

t=30
n=2880
percent=90/V_heater
number=4
if V_heater<90
   heat_loss=0.0084*V_heater+0.4
else
   heat_loss=0.0048*V_heater+0.72
end

Elec=P_elec*t
V_mix=percent*V_heater
D_heater=0.6;
H_thermo=0.2
V_ther=1000*pi*(D_heater^2)*H_thermo/4
V_stra=(1-percent)*V_heater
loss=heat_loss*percent*t/24
loss_ther=heat_loss*V_ther*t/(V_heater*24)
loss_stra=heat_loss*t/24-loss-loss_ther
temp_switchon=58
temp_setup=60

%Load the input file:
[datacy, headertext]=xlsread(inputfile)

%initial values
temp_hstd=temp_setup
temp_ststd=temp_setup
temp_hcstd=temp_setup
temp_stcstd=temp_setup
sum_Q=0
sum_E=0
sum_pump=0
sum_elec=0
ssum_elec=0
V_pre=V_ther
VV_pre=V_ther
Elec_pre=0
```

```
EElec_pre=0
Elec_addpre=0
sum_elecadd=0
EElec_addpre=0
ssum_elecadd=0
temp_supply=63

%Loop to read input files
for ii=1:n
   date=datacy(ii,1)
   time=datacy(ii,2)
   flow_cold=datacy(ii,3)
   temp_cold=datacy(ii,4)
   temp_cstd=datacy(ii,5)
   temp_cmix=datacy(ii,6)
   out1(ii,1)=datacy(ii,1)
   out2(ii,1)=datacy(ii,2)

   % PART 2: HOT WATER CYLINDER CALCULATION
   clear T cp rho mu nu k alpha beta Pr
   T=(temp_ststd+temp_cstd)/2
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
   cp_avg=cp
   rho_avg=rho
   clear T cp rho mu nu k alpha beta Pr
   T=temp_hstd
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
   cp_hstd=cp
   rho_hstd=rho
   clear T cp rho mu nu k alpha beta Pr
   T=temp_ststd
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
   cp_ststd=cp
   rho_ststd=rho
   clear T cp rho mu nu k alpha beta Pr
   T=(temp_stcstd+temp_cold)/2
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
   cp_cavg=cp
   rho_cavg=rho
   clear T cp rho mu nu k alpha beta Pr
   T=temp_hcstd
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
   cp_hcstd=cp
   rho_hcstd=rho
   fprintf('temp_hcstd=%8.2f\n',temp_hcstd)
   clear T cp rho mu nu k alpha beta Pr
```

```
T=temp_stcstd
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
cp_stcstd=cp
rho_stcstd=rho

if flow_cold<=0
   flow_cold=0
   %condition 1:Heat exhanger is installed
   clear temp_hout temp_stout Elec_used delta_E1 delta_E2 delta_E E3 V
   if temp_hstd>=temp_switchon
      if Elec_pre<=0
         temp_hout=temp_hstd-loss_ther/(V_ther*rho_hstd*cp_hstd*0.001)
         temp_stout=temp_hout
         V=V_ther
         Elec_used=0
      else
         if temp_ststd>=temp_setup
            temp_hout=temp_hstd-loss_ther/(V_ther*rho_hstd*cp_hstd*0.001)
            temp_stout=temp_hout
            V=V_ther
            Elec_used=0
         else
            delta_E1=V_pre*0.001*rho_hstd*cp_hstd*(temp_ststd-temp_hstd)
            delta_E1=max(delta_E1,0)
            delta_E2=V_stra*0.001*rho_hstd*cp_hstd*(temp_setup-temp_ststd)
            delta_E=delta_E1+delta_E2+loss_ther+loss_stra
            fprintf('delta_E1=%8.4f\n',delta_E1)
            fprintf('delta_E2=%8.4f\n',delta_E2)
            fprintf('delta_E=%8.4f\n',delta_E)

            if Elec>=delta_E
               temp_hout=temp_setup
               temp_stout=temp_setup
               V=V_ther
               Elec_used=delta_E
            elseif ((Elec>=delta_E1+loss_ther+loss_stra) & (Elec<delta_E))
               temp_stout=temp_ststd+(Elec-delta_E1-loss_ther-loss_stra)/
               (V_stra*0.001*rho_ststd*cp_ststd)
               temp_hout=temp_stout
               V=V_ther
               Elec_used=Elec
            else
               Elec_used=Elec
               E3=Elec-loss_ther-loss_stra
               temp_hout=temp_hstd+E3/(V_pre*0.001*rho_hstd*cp_hstd)
               if V_pre<V_stra
```

```
                temp_stout=temp_ststd-(loss_ther+loss_stra)*(V_stra-V_pre)/
                (V_stra*(V_stra-V_pre)*0.001*rho_ststd*cp_ststd)
                V=min(V_pre,V_stra)
            else
                temp_stout=temp_hout
                V=V_stra
            end
        end
    end  %for 'if Elec>=delta_E'
  end
else

  if  Elec_pre<=0
    delta_E=V_stra*0.001*rho_hstd*cp_hstd*(temp_setup- temp_hstd)
    +loss_ther+ loss_stra
    fprintf('delta_E=%8.4f\n',delta_E)
    Elec_used=min(Elec,delta_E)
    temp_hout=temp_hstd+(Elec_used-loss_ther-loss_stra)/(V_stra*rho_hstd*
    cp_hstd*0.001)
    temp_stout=temp_hout
    V=V_stra
  else
    if  temp_ststd>=temp_setup
      temp_hout=temp_hstd-loss_ther/(V_ther*rho_hstd*cp_hstd*0.001)
      temp_stout=temp_hout
      V=V_ther
      Elec_used=0
    else
      delta_E1=V_pre*0.001*rho_hstd*cp_hstd*(temp_ststd-temp_hstd)
      delta_E2=V_stra*0.001*rho_hstd*cp_hstd*(temp_setup-temp_ststd)
      delta_E=delta_E1+delta_E2+loss_ther+loss_stra

      if  Elec>=delta_E
        temp_hout=temp_setup
        temp_stout=temp_setup
        V=V_ther
        Elec_used=delta_E
      elseif ((Elec>=delta_E1+loss_ther+loss_stra) & (Elec<delta_E))
        temp_stout=temp_ststd+(Elec-delta_E1-loss_ther-loss_stra)/
        (V_stra*0.001*rho_ststd*cp_ststd)
        temp_hout=temp_stout
        V=V_stra
        Elec_used=Elec
      else
        Elec_used=Elec
        E3=Elec-loss_ther-loss_stra
```

```
                temp_hout=temp_hstd+E3/(V_pre*0.001*rho_hstd*cp_hstd)
                if V_pre<V_stra
                    temp_stout=temp_ststd-(loss_ther+loss_stra)*(V_stra-V_pre)/(V_stra*(V_stra-V_pre)*0.001*rho_ststd*cp_ststd)
                    V=min(V_pre,V_stra)
                else
                    temp_stout=temp_hout
                    V=V_stra
                end
            end
        end
    end
end

%condition 2:Heat exhanger is not installed;
clear temp_hcout temp_stcout EElec_used delta_EE1 delta_EE2 delta_EE EE3 VV

if temp_hcstd>=temp_switchon
    if EElec_pre<=0
        temp_hcout=temp_hcstd-loss_ther/(V_ther*rho_hcstd*cp_hcstd*0.001)
        temp_stcout=temp_hcout
        VV=V_ther
        EElec_used=0
    else
        if temp_stcstd>=temp_setup
            temp_hcout=temp_hcstd-loss_ther/(V_ther*rho_hcstd*cp_hcstd*0.001)
            temp_stcout=temp_hcout
            VV=V_ther
            EElec_used=0
        else
            delta_EE1=VV_pre*0.001*rho_hcstd*cp_hcstd*(temp_stcstd-temp_hcstd)
            delta_EE1=max(delta_EE1,0)
            delta_EE2=V_stra*0.001*rho_hcstd*cp_hcstd*(temp_setup-temp_stcstd)
            delta_EE=delta_EE1+delta_EE2+loss_ther+loss_stra
            fprintf('delta_EE1=%8.4f\n',delta_EE1)
            fprintf('delta_EE2=%8.4f\n',delta_EE2)
            fprintf('delta_EE=%8.4f\n',delta_EE)
            if Elec>=delta_EE
                temp_hcout=temp_setup
                temp_stcout=temp_setup
                VV=V_ther
                EElec_used=delta_EE
            elseif ((Elec>=delta_EE1+loss_ther+loss_stra) & (Elec<delta_EE))
                temp_stcout=temp_stcstd+(Elec-delta_EE1-loss_ther-loss_stra)/(V_stra*0.001*rho_stcstd*cp_stcstd)
```

```
                temp_hcout=temp_stcout
                VV=V_ther
                EElec_used=Elec
            else
                Elec<delta_EE1
                EElec_used=Elec
                EE3=Elec-loss_ther-loss_stra;
                temp_hcout=temp_hcstd+EE3/(VV_pre*0.001*rho_hcstd*cp_hcstd)
                if VV_pre<V_stra
                    temp_stcout=temp_stcstd-(loss_ther+loss_stra)*(V_stra-
                    VV_pre)/(V_stra*(V_stra-VV_pre)*0.001*rho_stcstd*cp_stcstd)
                    VV=min(VV_pre,V_stra)
                else
                    temp_stcout=temp_hcout
                    VV=V_stra
                end
            end
        end
    end
else
    if EElec_pre<=0
        delta_EE=V_stra*0.001*rho_hcstd*cp_hcstd*(temp_setup-temp_hcstd)
        +loss_ther+loss_stra
        fprintf('delta_EE=%8.4f\n',delta_EE)
        EElec_used=min(Elec,delta_EE)
        temp_hcout=temp_hcstd+(EElec_used-loss_ther-loss_stra)/
        (V_stra*rho_hcstd*cp_hcstd*0.001)
        temp_stcout=temp_hcout
        VV=V_stra
    else
        if temp_stcstd>=temp_setup
            temp_hcout=temp_hcstd-loss_ther/(V_ther*rho_hcstd*cp_hcstd*0.001)
            temp_stcout=temp_hcout
            VV=V_ther
            EElec_used=0
        else
            delta_EE1=VV_pre*0.001*rho_hcstd*cp_hcstd*(temp_stcstd
            -temp_hcstd)
            delta_EE2=V_stra*0.001*rho_hcstd*cp_hcstd*(temp_setup-temp_stcstd)
            delta_EE=delta_EE1+delta_EE2+loss_ther+loss_stra

            if Elec>=delta_EE
                temp_hcout=temp_setup
                temp_stcout=temp_setup
                VV=V_ther
                EElec_used=delta_EE
```

```
                elseif ((Elec>=delta_EE1+loss_ther+loss_stra) & (Elec<delta_EE))
                    temp_stcout=temp_stcstd+(Elec-delta_EE1-loss_ther
                    -loss_stra)/(V_stra*0.001*rho_stcstd*cp_stcstd)
                    temp_hcout=temp_stcout
                    VV=V_stra
                    EElec_used=Elec
                else
                    EElec_used=Elec
                    EE3=Elec-loss_ther-loss_stra;
                    temp_hcout=temp_hcstd+EE3/(VV_pre*0.001*rho_hcstd*cp_hcstd);
                    if VV_pre<V_stra
                        temp_stcout=temp_stcstd-(loss_ther+loss_stra)*(V_stra
                        -VV_pre)/(V_stra*(V_stra-VV_pre)*0.001*rho_stcstd*cp_stcstd)
                        VV=min(VV_pre,V_stra);
                    else
                        temp_stcout=temp_hcout;
                        VV=V_stra;
                    end
                end
            end
        end
    end
    Elec_add=0
    EElec_add=0
end

if flow_cold>0
            %condition 1:Heat exhanger is installed
    clear temp_hout temp_stout Elec_used delta_E1 delta_E2 delta_E E3 V

    if temp_hstd>=temp_switchon
        if Elec_pre<=0
            temp_hout=(temp_hstd*0.001*V_pre*rho_hstd*cp_hstd*flow_cold*t
            +temp_cstd*0.001*rho_avg*cp_avg- loss_ther*V_pre/V_ther)/
            (0.001*(V_pre*rho_hstd*cp_hstd+flow_cold*t*rho_avg*cp_avg))

            if V_pre<V_stra
                temp_stout=temp_ststd-(loss_ther+loss_stra)*(V_stra-V_pre)/
                (V_stra*(V_stra-V_pre)*0.001*rho_ststd*cp_ststd)
                V=min(V_pre,V_stra)
            else
                temp_stout=temp_hout
                V=V_stra
            end
            Elec_used=0
        else
```

```
    if temp_ststd>=temp_setup
      temp_hout=temp_hstd-loss_ther/(V_ther*rho_hstd*cp_hstd*0.001)
      temp_stout=temp_hout
      V=V_ther
      Elec_used=0
    else
      delta_E1=flow_cold*0.001*t*rho_avg*cp_avg*(temp_ststd-temp_cstd)
      +V_pre*0.001*rho_hstd*cp_hstd*(temp_ststd-temp_hstd)
      delta_E1=max(delta_E1,0)
      delta_E2=(flow_cold*0.001*t+V_stra*0.001)*rho_ststd*cp_ststd
      *(temp_setup-temp_ststd)
      delta_E=delta_E1+delta_E2+loss_ther+loss_stra
      if Elec>=delta_E
        temp_hout=temp_setup
        temp_stout=temp_setup
        V=V_ther
        Elec_used=delta_E
      elseif ((Elec>=delta_E1+loss_ther+loss_stra) & (Elec<delta_E))
        temp_stout=temp_ststd+(Elec-delta_E1-loss_ther-loss_stra)
        /((V_stra+flow_cold*t)*0.001*rho_ststd*cp_ststd)
        temp_hout=temp_stout
        V=V_ther
        Elec_used=Elec
      else
        Elec_used=Elec
        E3=Elec-loss_ther-loss_stra;
        temp_hout=(E3+flow_cold*t*0.001*rho_avg*cp_avg*temp_cstd
        +V_pre*0.001*rho_hstd*cp_hstd*temp_hstd)/(flow_cold*t*0.001
        *rho_avg*cp_avg+V_pre*0.001*rho_hstd*cp_hstd)

        if V_pre<V_stra
          temp_stout=temp_ststd-(loss_ther+loss_stra)*(V_stra-V_pre)
          /(V_stra*((V_stra-V_pre)+flow_cold*t)*0.001*rho_ststd*cp_ststd)
          V=V_pre+flow_cold*t
          V=min(V,V_stra)
        else
          temp_stout=temp_hout
          V=V_stra
        end
      end
    end
    fprintf('temp_hout=%8.4f\n',temp_hout);
    fprintf('temp_stout=%8.4f\n',temp_stout);
  end
else
  delta_E1=flow_cold*0.001*t*rho_avg*cp_avg*(temp_ststd-temp_cstd)
```

```
        +V_pre*0.001* rho_hstd*cp_hstd*(temp_ststd-temp_hstd)
        delta_E2=(flow_cold*0.001*t+V_stra*0.001)*rho_ststd*cp_ststd
        *(temp_setup-temp_ststd)
        delta_E=delta_E1+delta_E2+loss_ther+loss_stra;

        if  Elec>=delta_E
          temp_hout=temp_setup
          temp_stout=temp_setup
          V=V_ther
          Elec_used=delta_E
        elseif ((Elec>=delta_E1+loss_ther+loss_stra) & (Elec<delta_E))
          temp_stout=temp_ststd+(Elec-delta_E1-loss_ther-loss_stra)
          /((V_stra+flow_cold*t)*0.001*rho_ststd*cp_ststd)
          temp_hout=temp_stout
          V=V_ther
          Elec_used=Elec
        else
          Elec_used=Elec
          E3=Elec-loss_ther-loss_stra
          temp_hout=(E3+flow_cold*t*0.001*rho_avg*cp_avg*temp_cstd
          +V_pre*0.001*rho_hstd*cp_hstd*temp_hstd)/(flow_cold*t*0.001*rho_avg
          *cp_avg+V_pre*0.001*rho_hstd*cp_hstd)

          if  V_pre<V_stra
            temp_stout=temp_ststd-(loss_ther+loss_stra)*(V_stra-V_pre)/(V_stra
            * ((V_stra-V_pre)+flow_cold*t)*0.001*rho_ststd*cp_ststd)
            V=V_pre+flow_cold*t
            V=min(V,V_stra)
          else
            temp_stout=temp_hout
            V=V_stra
          end
        end %
        fprintf('temp_hout=%8.4f\n',temp_hout);
        fprintf('temp_stout=%8.4f\n',temp_stout);
      end

      %Condition 2
      clear temp_hcout temp_stcout EElec_used delta_EE1 delta_EE2 delta_EE EE3
VV
      if  temp_hcstd>=temp_switchon
        if  EElec_pre<=0
          temp_hcout=(temp_hcstd*0.001*VV_pre*rho_hcstd*cp_hcstd
          +temp_cold*0.001*flow_cold*t*rho_cavg*cp_cavg-loss_ther
          *VV_pre/V_ther)/(0.001*(VV_pre*rho_hcstd*cp_hcstd+flow_cold
          *t*rho_cavg*cp_cavg))
```

```
        if VV_pre<V_stra
            temp_stcout=temp_stcstd-(loss_ther+loss_stra)*(V_stra-VV_pre)
            /(V_stra*(V_stra-VV_pre)*0.001*rho_stcstd*cp_stcstd)
            VV=min(VV_pre,V_stra)
        else
            temp_stcout=temp_hcout
            VV=V_stra
        end
        EElec_used=0
    else
        if temp_stcstd>=temp_setup
            temp_hcout=temp_hcstd-loss_ther/(V_ther*rho_hcstd*cp_hcstd*0.001)
            temp_stcout=temp_hcout
            VV=V_ther
            EElec_used=0
        else
            delta_EE1=flow_cold*0.001*t*rho_cavg*cp_cavg*(temp_stcstd
            -temp_cold)+VV_pre*0.001*rho_hcstd*cp_hcstd*(temp_stcstd-
            temp_hcstd)
            delta_EE1=max(delta_EE1,0)
            delta_EE2=(flow_cold*0.001*t+V_stra*0.001)*rho_stcstd*cp_stcstd
            *(temp_setup-temp_stcstd)
            delta_EE=delta_EE1+delta_EE2+loss_ther+loss_stra

            if Elec>=delta_EE
                temp_hcout=temp_setup
                temp_stcout=temp_setup
                VV=V_ther
                EElec_used=delta_EE
            elseif ((Elec>=delta_EE1+loss_ther+loss_stra) & (Elec<delta_EE))
                temp_stcout=temp_stcstd+(Elec-delta_EE1-loss_ther-loss_stra)
                / ((V_stra+flow_cold*t)*0.001*rho_stcstd*cp_stcstd)
                temp_hcout=temp_stcout
                VV=V_ther
                EElec_used=Elec
            else
                EElec_used=Elec
                EE3=Elec-loss_ther-loss_stra
                temp_hcout=(EE3+flow_cold*t*0.001*rho_cavg*cp_cavg*temp_cold
                +VV_pre*0.001*rho_hcstd*cp_hcstd*temp_hcstd)/(flow_cold*t
                *0.001*rho_cavg*cp_cavg+VV_pre*0.001*rho_hcstd*cp_hcstd)

                if VV_pre<V_stra
                    temp_stcout=temp_stcstd-(loss_ther+loss_stra)*(V_stra-VV_pre)
                    /(V_stra*((V_stra-VV_pre)+flow_cold*t)*
```

```
                0.001*rho_stcstd*cp_stcstd)
                VV=VV_pre+flow_cold*t
                VV=min(VV,V_stra)
              else
                temp_stcout=temp_hcout
                VV=V_stra
              end
            end
          end
        end
      else

        delta_EE1=flow_cold*0.001*t*rho_cavg*cp_cavg*(temp_stcstd
        -temp_cold)+VV_pre*0.001*rho_hcstd*cp_hcstd*(temp_stcstd-temp_hcstd)
        delta_EE2=(flow_cold*0.001*t+V_stra*0.001)*rho_stcstd*
        cp_stcstd*(temp_setup-temp_stcstd)
        delta_EE=delta_EE1+delta_EE2+loss_ther+loss_stra
        if Elec>=delta_EE
          temp_hcout=temp_setup
          temp_stcout=temp_setup
          VV=V_ther
          EElec_used=delta_EE
        elseif ((Elec>=delta_EE1+loss_ther+loss_stra) & (Elec<delta_EE))
          temp_stcout=temp_stcstd+(Elec-delta_EE1-loss_ther-loss_stra)
          /((V_stra+flow_cold*t)*0.001*rho_stcstd*cp_stcstd)
          temp_hcout=temp_stcout
          VV=V_ther
          EElec_used=Elec
        else
          EElec_used=Elec
          EE3=Elec-loss_ther-loss_stra
          temp_hcout=(EE3+flow_cold*t*0.001*rho_cavg*cp_cavg*temp_cold
          +VV_pre*0.001*rho_hcstd*cp_hcstd*temp_hcstd)/(flow_cold*t*0.001
          *rho_cavg*cp_cavg+VV_pre*0.001*rho_hcstd*cp_hcstd)

          if VV_pre<V_stra
            temp_stcout=temp_stcstd-(loss_ther+loss_stra)*(V_stra-VV_pre)
            / (V_stra*((V_stra-VV_pre)+flow_cold*t)*0.001*rho_stcstd*cp_stcstd)
            VV=VV_pre+flow_cold*t;
            VV=min(VV,V_stra)
          else
            temp_stcout=temp_hcout
            VV=V_stra
          end
        end
      end
```

```
    %Condition 1:
    clear T cp rho mu nu k alpha beta Pr
    T=(temp_supply+temp_stout)/2
    [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
    cp_avgsupply=cp
    rho_avgsupply=rho
    clear T cp rho mu nu k alpha beta Pr
    T=(temp_supply+temp_stcout)/2
    [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
    cp_cavgsupply=cp
    rho_cavgsupply=rho

    Elec_add=flow_cold*0.001*t*cp_avgsupply*rho_avgsupply*(temp_supply-temp_stout)
    EElec_add=flow_cold*0.001*t*cp_cavgsupply*rho_cavgsupply*(temp_supply-temp_stcout)
end

%Condition 1:
temp_add=temp_supply
temp_cadd=temp_supply
Elec_addmax=max(Elec_add,Elec_addpre)
P_addmax=Elec_addmax/t
    sum_elecadd=sum_elecadd+Elec_add
sum_elec=sum_elec+Elec_used
out18(ii,1)=Elec_used/3600
out19(ii,1)=sum_elec/3600
out20(ii,1)=temp_hstd
out21(ii,1)=temp_hout
out22(ii,1)=temp_ststd
out23(ii,1)=temp_stout
out24(ii,1)=V
out241(ii,1)=V_pre
out32(ii,1)=temp_add
out33(ii,1)=Elec_add/3600
out34(ii,1)=P_addmax
out35(ii,1)=sum_elecadd/3600

temp_hstd=temp_hout
temp_ststd=temp_stout
Elec_pre=Elec_used
V_pre=V
Elec_addpre=Elec_addmax

%Condition 2:
```

```
    EElec_addmax=max(EElec_add,EElec_addpre)
    P_caddmax=EElec_addmax/t
        ssum_elecadd=ssum_elecadd+EElec_add
        ssum_elec=ssum_elec+EElec_used
    out25(ii,1)=EElec_used/3600
    out26(ii,1)=ssum_elec/3600
    out27(ii,1)=temp_hcstd
    out28(ii,1)=temp_hcout
    out29(ii,1)=temp_stcstd
    out30(ii,1)=temp_stcout
    out31(ii,1)=VV
    out311(ii,1)=VV_pre
    out36(ii,1)=temp_cadd
    out37(ii,1)=EElec_add/3600
    out38(ii,1)=P_caddmax
    out39(ii,1)=ssum_elecadd/3600

    temp_hcstd=temp_hcout
    temp_stcstd=temp_stcout
    EElec_pre=EElec_used
    VV_pre=VV
    EElec_addpre=EElec_addmax
    result6=[out18,out19,out20,out21,out22,out23,out24,out241]
    result7=[out25,out26,out27,out28,out29,out30,out31,out311]
    result8=[out32,out33,out34,out35,out36,out37,out38,out39]
  end
```

B.3 SUB-PROGRAMMES

B.3.1 Overall Heat Transfer Coefficient

```
%Funtion
function u=overcoeffi(flow_cent,delta_temp)

flows=1:1:4
x=[0 5 15 24 40]
y=[0 275 490 620 705
  0 420 680 800 840
  0 500 750 840 905
  0 595 785 860 935]
u=interp2(x,flows,y,delta_temp,flow_cent,'cubic')
fprintf('delta_temp=%6.2f\n',delta_temp)
fprintf('flow_cent=%6.2f\n',flow_cent)
fprintf('overall coefficient u=%6.1f\n',u)
```

B.3.2 Mixing Process in the Mixing Chamber of the Hot Water Tank

```
%Function
function [temp_cmix]=cmix(t,temp_cent,temp_cout,temp_cold,temp_cstd,
flow_cent,flow_cold, V_mix,loss)

%calculate cp & rho
clear T cp rho mu nu k alpha beta Pr
T=temp_cstd;
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
cp_cstd=cp
rho_cstd=rho

clear T cp rho mu nu k alpha beta Pr
T=temp_cent;
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
cp_cent=cp
rho_cent=rho

clear T cp rho mu nu k alpha beta Pr
T=temp_cold
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
cp_cold=cp
rho_cold=rho

clear T cp rho mu nu k alpha beta Pr
T=temp_cout
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
cp_cout=cp
rho_cout=rho

Temp_cmix=temp_cstd
rho_cmix=rho_cstd
cp_cmix=cp_cstd
```

```
for jj=1:10
   E1=V_mix*rho_cstd*cp_cstd*temp_cstd*0.001
   E2=t*flow_cent*0.001*(rho_cout*cp_cout*temp_cout-rho_cent*cp_cent*temp_cent)/60
   E3=t*flow_cold*0.001*rho_cold*cp_cold*temp_cold

   Heatloss=loss
   rhocpt=V_mix*0.001*rho_cmix*cp_cmix+t*flow_cold*0.001*rho_cmix*cp_cmix
   temp_cmix=(E1+E2+E3-Heatloss)/rhocpt
   temp_cmix=max(temp_cmix,temp_cold)
   delta_temp=abs(Temp_cmix-temp_cstd)
   if  delta_temp>0.1
      Temp_cmix=temp_cmix
   else
      clear T cp rho mu nu k alpha beta Pr
      T=temp_cmix
      [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T)
      cp_cmix=cp
      rho_cmix=rho
      jj=10
   end
end
```

B.3.3 Property

```
%Function
function [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T);

%Define variables:
% T    --temperature, degree centigrade;
% rho  --mass density, kg/m^3;
% cp   --specific heat at constant pressure, kJ/kg.K;
% k    --thermal conductivity, W/m.K;

%Calculate properties:
rho=-4.516E-3*T^2-0.01224*T+1000;
cp=-2.686E-7*T^3+4.589E-5*T^2-2.214E-3*T+4.211;
k=-9.04E-6*T^2+2.094E-3*T+0.5621;
```

B.3.4 Waste Water Calculation

```
function [cp_wstd,rho_wstd,temp_went,cp_went,rho_went,tempin_wstd,
        cpin_wstd,rhoin_wstd]=mixed(t,V_water,temp_wstd,temp_went,flow_went)
%calculate cp & rho
clear T cp rho mu nu k alpha beta Pr
T=temp_wstd;
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T);
cp_wstd=cp;
rho_wstd=rho;
clear T cp rho mu nu k alpha beta Pr;
T=temp_went;
[rho,cp,mu,nu,k,alpha,beta,Pr]=property(T);
cp_went=cp;
rho_went=rho;
Tin_wstd=(temp_wstd+temp_went)/2;
for jj=1:10
   clear T cp rho mu nu k alpha beta Pr;
   T=Tin_wstd;
   [rho,cp,mu,nu,k,alpha,beta,Pr]=property(T);
   cpin_wstd=cp;
   rhoin_wstd=rho;
   tempin_wstd=(V_water*rho_wstd*cp_wstd*temp_wstd+flow_went*t
               *0.001*rho_went*cp_went*temp_went)/((V_water*rho_wstd
               +flow_went*t*0.001*rhoin_wstd)*cpin_wstd);
   %Compare
   delta_temp=abs(Tin_wstd-tempin_wstd);
   if delta_temp>0.1
      Tin_wmix=tempin_wstd;
   else
      break
   end
end
```